Marine Engine Indicating

Charles S. Linch, M. E.

OCEAN TUG "WILLIAM M. MILLS"

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Marine Engine Indicating

A Complete Treatise
on the Indicator and Indicator Diagrams
as applied to Marine Engines

BY

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Consulting and Constructing Naval Architect
and Marine Engineer

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BOSTON
American Steam Gauge & Valve Mfg. Co.
Camden Street

CONTENTS

THIS work is respectfully dedicated to my friend, R. B. Phillips, Treasurer and Manager of the American Steam Gauge & Valve Manufacturing Company, through whose Indicator, the American-Thompson, I have been able in all my professional work to accomplish most perfect results, and because it is my unqualified opinion that the facility and accuracy of this instrument is unequaled.

The importance of a perfect instrument in the expert work which I am constantly called upon to perform has compelled me to make this selection by thorough tests and the absence of all prejudice.

It is, therefore, in this same spirit that I give credit where credit is due.

CHARLES S. LINCH.

FOREWORD

It has been the writer's observation — and doubtless the reader's as well — that text books written on the subject of indicators are invariably based on experiences with stationary engines.

That a thorough treatise on this all-important device, with special reference to its application to marine engines is greatly needed, is obvious to every marine engineer, and this work is undertaken expressly to meet that need, particular care being exercised, especially in all the analyses of diagrams, to be lucid and concise, rather than elaborately technical.

The history of the indicator has been purposely avoided, as being superfluous, the writer deeming it of far greater importance to confine himself especially to a complete description of the most accurate of the modern type.

In the analysis of diagrams it is important, when adjustment of valves must be made, to be able to construct and discuss the valve diagrams, and the object here has been to explain the methods in a clear manner, eliminating all geometrical proof.

All diagrams shown were taken, in actual practice, from modern marine engines.

If by writing this work I have been of help to those who are seeking this knowledge, I shall feel amply repaid.

I am greatly indebted to Mr. Harry Vansciver, Division Superintendent, Merchants and Miners Transportation Company, for the analysis of the steamship "Tuscan."

THE AUTHOR.

MARINE INDICATING

CHAPTER I

THE steam engine indicator is an instrument which, through the proper functioning of its various parts, depicts upon paper a diagram which should accurately represent the various changes of pressure on one side of the piston of the steam engine during both the forward and return strokes.

Not only does the diagram show these variations, but it shows defects of design and adjustment, enabling the engineer to rectify faulty adjustment, and to determine any changes which would be conducive to increased economy and efficiency.

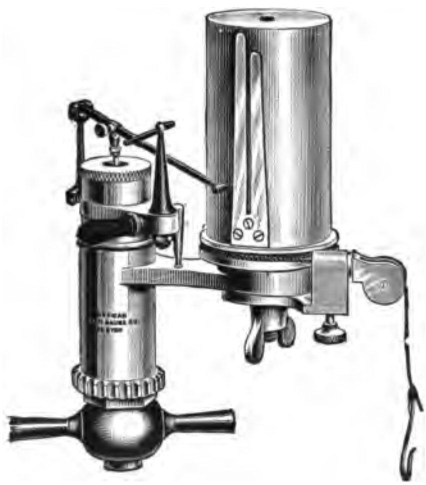


Fig. 1

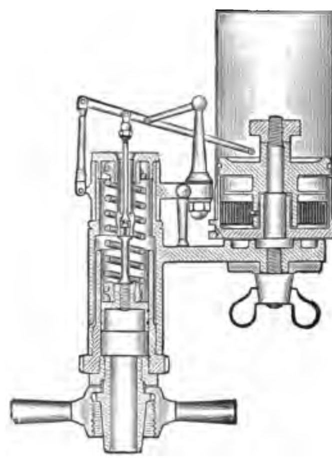


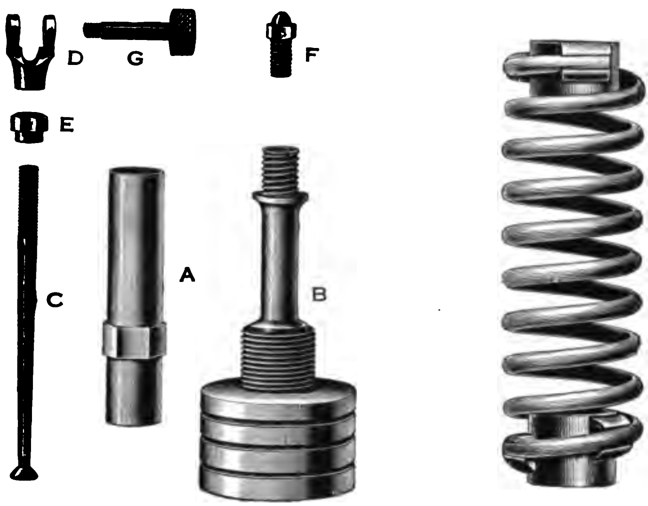
Fig. 2

Fig. 1 shows an outside view and Fig. 2 a section through the incased spring instrument manufactured by the American Steam Gauge and Valve Manufacturing Company of Boston, Mass., known to the engineering profession as the American Thompson Improved Indicator. This instrument consists of an outer cylinder or casing into which is secured the liner in which the piston travels. This liner is made of a special hard bronze composition, which differs slightly from the composition of which the piston is made. The object of having the liner and piston made of different compositions is to obtain a uniform expansion. The space between the outer casing and liner forms a suitable steam jacket. The bracket which carries the paper drum

spindle and the casing are one casting. This bracket is of sufficient dimensions to form a very rigid and strong appendage, the distance between the center of cylinder and center of drum spindle being only sufficient to insure the pencil striking the proper position on the paper drum in a vertical plane. The pencil motion being three to one, this distance is therefore such that danger of bending with the light construction is eliminated.

The spindle is of steel and, as will be observed, is screwed into the bracket and shouldered; the end extending through the bracket carries the guide pulley bracket and wing nut.

The bearing surface for the paper drum pulley is large, insuring ample bearing surface.



Piston

Fig. 2a

The piston Fig. 2*a* is of a special composition permitting a light construction yet possessing the requisite strength to prevent expansion from pressure, and is grooved for water packing.

The stem of the piston is constructed throughout of steel; the upper part consists of the sleeve "A" which acts as a guide passing through the cylinder cap. The piston "B" is connected with the pencil lever by a connecting rod "C" having a cross-head "D" at the upper end, which acts as a yoke, making connection with pencil lever by knurled-headed screw "G" connecting yoke with lever.

The cross-head is held in place by a small hexagonal lock nut "E." The top of the connecting rod is threaded, permitting the raising or lowering of the cross-head, thus securing adjustment of the atmospheric line on the diagram.

The lower end of the connecting rod forms a socket which rests on a ball stud "F," which, in turn, is adjustable in the piston stem. The result is a perfect ball and socket joint, and provides means for taking up any lost motion.

The parallel motion is made of drop-forged, compressed steel, and is carried on a sleeve, which is fitted to the upper end of the steam cylinder, being held in place by the milled cylinder cap. The pencil lever has a vertical motion in the ratio of three to one, and is guided by a short connecting link, which vibrates about a pin carried by the post. The post is carried by an arm cast with the sleeve. A link connecting the pencil lever and vibrating about a center carried also on the sleeve, acts as a fulcrum. The yoke as mentioned connects the piston with the pencil lever.

This construction insures an absolute straight line for pressure line; any inclination of this line in any diagram can therefore be attributed to other causes.

The end of the pencil lever is split, thus forming a spring sleeve to take the lead or German silver points.

Through the arm of the sleeve there is drilled and tapped a hole for the adjusting screw, as shown.

On the bracket carrying the paper drum there is fitted a stop to prevent injury to pencil lever, by introducing excessive friction on card, from too great pressure of lead against paper. The sleeve being free to turn, the adjustment of adjusting screw determines the pressure put on pencil.

The connection of the indicator to the straight or three-way cocks is through the medium of a swivel coupling, having a tailpiece which is secured into the lower end of the cylinder. This tailpiece is provided with a shoulder against which the inner flange of the coupling proper rests; this forms a perfect swivel coupling and is a decided improvement over those having right and left hand thread.

Springs

The springs are made of the finest quality steel wire, and are wound on a mandrel and tempered in the most scientific manner. This mandrel on which all springs are wound is from four to four and one-half threads per inch. In the springs furnished with these instruments there is therefore more wire to each spring, and hence less strain than if wound on mandrels of two and three threads per inch. The heads of the springs are of brass, drilled and tapped to receive the piston and cylinder cap.

In securing the heads to the spring, no solder is used. The cut (Fig. 2*a*) shows clearly the construction.

Paper Drum

The paper drum is of brass tubing, turned true, faced, capped and bored for pulley, and is light, yet possessing requisite strength.

The tension spring is carried by the drum pulley, the spring case forming an integral part of same. The tension of the spring is adjusted by turning the knurled cap, the cap is prevented from slipping by friction of the knurled lock nut. The construction is clearly shown in Fig. 3.

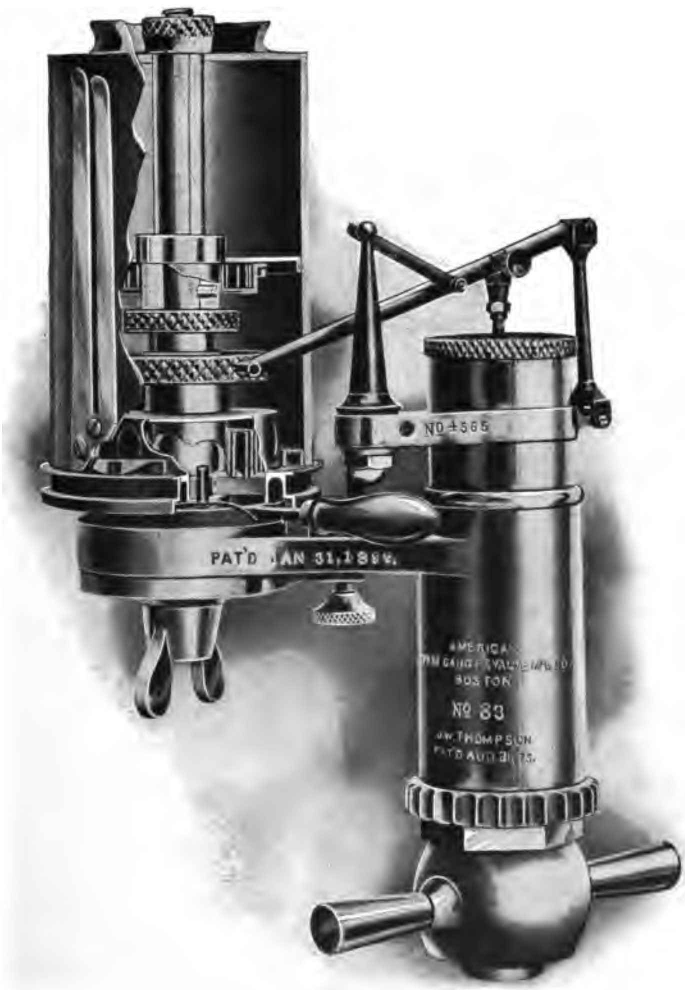


Fig. 3

Fig. 6 shows a section through the paper drum of an instrument fitted with detent motion.

Leading Pulley

The leading pulley shown in Fig. 4 consists of a wheel which is carried on an adjustable bearing. This bearing as shown is carried by a stand which is cast with a palm, the palm is drilled so that it can pass over the extension of the paper drum spindle. This palm is clamped by the wing nut as shown in Figs. 1 and 2.

The cord from the grooved wheel of paper drum is passed through the hole in the pulley sleeve, thence passing over the pulley to the driving cord from reducing motion. After the leading pulley is adjusted it is clamped by the knurled head screw as shown. It will be noted that the cord from paper cylinder is always tangent to the groove in leading pulley.

Fig. 4

Detent Motion

Fig. 5 shows the instrument fitted with detent motion, and Fig. 6 shows a section through the paper drum of this instrument. It will be noticed that in order to stop the paper cylinder it is only necessary to move lever "A" in the direction traveled by the paper cylinder until the cylinder releases itself. The cylinder will then remain stationary, at which time the completed diagram can be removed and a new card substituted. The lever must be returned to its original position.

Looking now at Fig. 3 we see that the pin which is carried by spring when in position as shown, drives the paper cylinder. This spring is drawn down when lever is pushed over, hence withdrawing pin, thus disengaging the paper drum from pulley. When lever is again thrown back, the spring is free to push pin into position as soon as the hole in drum and drum pulley coincide. Therefore, when new card has been put on drum, turn the milled rim "B" on top of drum forward until it catches. The drum will then be in gear, and hence will revolve in usual manner.

Exposed Spring Instrument

The exposed spring instrument shown in Fig. 6*a* is precisely the same as the incased spring instrument as far as construction and materials are concerned, except that the spring is not subject to variations of temperature and is visible at all times.

The parallel motion is the same as is used on the incased spring type, thus embodying this important feature which has made the American Thompson Improved Indicator so popular with the Engineering fraternity.

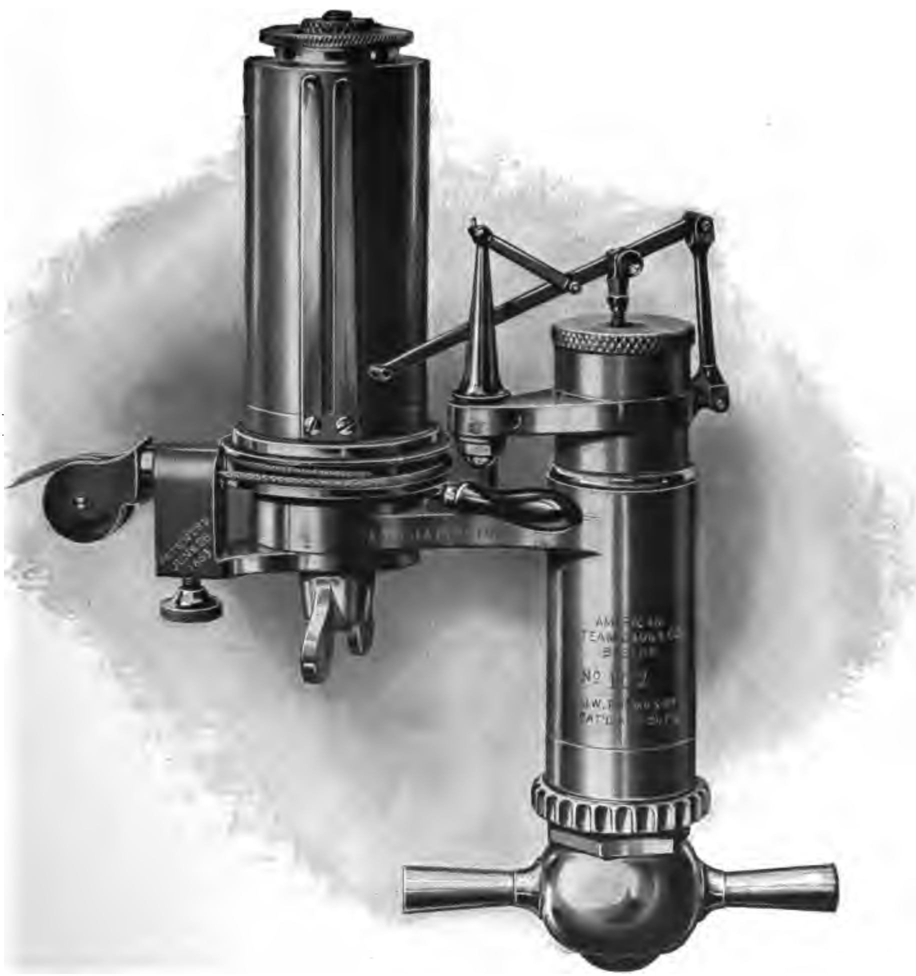


Fig 5.

The lower end of spring screws down to a shoulder located on piston above the cylinder. This shoulder is provided with four holes in which a pin is inserted for holding piston rod from turning when spring is to be inserted or removed.

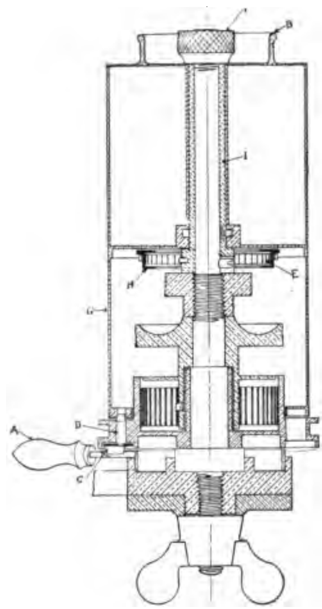


Fig. 6

Reducing Wheel

It frequently happens that engines are not fitted with reducing motions, and when such cases occur we must resort to the use of reducing wheels.

The reducing wheel shown in Fig. 7 is made of aluminum, brass, and steel combining lightness and strength, two very essential features. The wheel drum from which the cord passes to the cross-head arm or any other arrangement for driving, is $2\frac{3}{4}$ inches in diameter, and is made of aluminum. The coil spring for the take-up is in a separate case and connected by a three to one gear with the cord-wheel spindle, so that while the aluminum cord-wheel makes three revolutions, the spring makes but one. The spring can be adjusted to any desired tension to keep the cord taut on return stroke. The cord-wheel revolves on a steel screw, the thread of which has the same pitch as the cord, so that when the cord is drawn out the wheel travels as it revolves. Thus the cord is wound smoothly on the drum and passes straight through the guide pulley.

In using the reducing wheel on the indicator, remove the leading pulley (see Fig. 8) from the indicator and put the wheel on in place of it. Pass the drum cord around the small disk through the hole and under the holder. Observe that the cord is always wound round bushing or

Fig. 6a

disk from the left. Before hooking in, see that cord on wheel and indicator is taut at shortest part of stroke and that it will pull out a little further than longest part of stroke.

The cord from reducing wheel to cross-head must run in a straight line.

In unhooking the cord do not permit it to run unchecked but allow it to run slowly until the stop reaches the guide pulley.

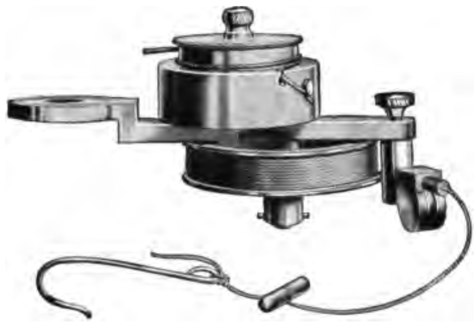


Fig. 7

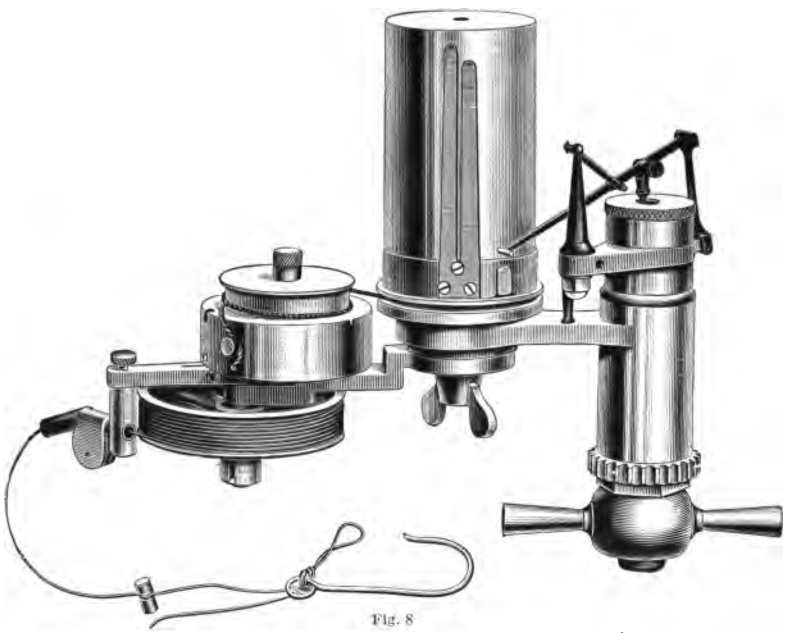


Fig. 8

Bushings are furnished of various sizes for small disks so that diagrams can be taken for any stroke up to 72″.

Having described the construction of the instrument we will now take up the subject of its care and adjustment.

Care

Before using an indicator take it apart and thoroughly clean and oil it. Starting at the steam cylinder, remove the small knurled-head screw connecting the pencil lever with the connecting rod. Unscrew the cylinder cap and withdraw the piston and parallel motion by holding the

instrument with one hand, and with thumb and finger lift up the sleeve. After the piston has been withdrawn, with one hand grasp the piston and with thumb and finger turn cylinder cap, unscrewing same from spring. Now unscrew spring from piston. Wipe out cylinder with clean waste, and see that all dirt, if any, is removed. Whilst the piston is out of instrument it is as well to look after the paper drum and its appendages.

Remove the knurled nut "F" (see Fig. 6); take off the paper drum, then with the wire clip (which is furnished with each instrument fitted with detent motion) remove the auxiliary spring case "H" by catching the end of the clip in the notches; then remove the spring and inner sleeve "I." After cleaning and oiling, replace the inner sleeve "I" by inserting it into the drum so that the pin on the outside of the sleeve will enter the slot inside of drum bearing and turn it until it comes to a stop; then with the wire clip catch hold of the auxiliary spring holder "H" and give the auxiliary spring "E" a tension of about ¼ turn, catching the points on the spring case "H" into the slots provided for them.

Whilst we have the auxiliary spring case and sleeve out it is necessary to be sure that the spindle is oiled; therefore, remove the lock nut, thus releasing the spring tension, then with screw driver (furnished with each instrument) remove the small screw on spindle, then remove lock nut, and lift off the paper drum pulley. Oil thoroughly and replace the pulley, and turn knurled cap, giving the spring the required tension and lock with lock nut; replace screw in spindle, thence replace paper drum, and finally the knurled nut "F."

Having selected the spring we wish to use, screw same to cylinder cap; next screw on the piston. Oil the piston with good cylinder oil and replace piston in cylinder; screw on the cylinder cap, and last, connect the pencil lever with connecting rod by inserting and gently screwing up the screw through yoke. Care must be exercised, and it is important to remember that the pencil lever must be disconnected first, and connected last. With the porpoise or watch oil (furnished with each instrument) oil the joints in the parallel motion. It is to be remembered that all parts of the instrument except the piston must not be oiled with any other oil except the kind furnished, and only a good cylinder oil is to be used on piston.

Adjustment

Great care must be exercised in adjusting the instrument. For the adjustment of the paper drum spring, the tension on this must not be greater than is absolutely required. To determine just what this should be in any case, we must, with the engine turning very slow, take a diagram; then with engine turning maximum number of evolutions,

take another diagram; with a pair of dividers measure the length of the diagrams; should the diagram taken with maximum turns show a difference in length the spring must be adjusted to give the same length. The tension on the spring will of course be greater for fast and less for slow turning engines, hence the necessity of adjusting to suit conditions.

The adjustment of the outside instrument is precisely the same as for incased spring.

The adjustment of the pencil is controlled by the adjusting screw, and should be such as to give as light a line consistent with clearness.

A diagram can very readily be distorted by excessive friction, and the data from same absolutely useless; besides the injury to the pencil lever.

After the instrument is removed from engine it should again be taken apart and all parts thoroughly cleaned and oiled, the cylinder thoroughly dried out, and all water of condensation removed from jacket. The springs should be thoroughly cleaned, dried, and oiled with porpoise oil. The piston should be oiled with porpoise oil when instrument is to be put away. All parts which are concealed, such as the ball and socket joint, should be wiped out by forcing a thin piece of linen down the sleeve with a toothpick, and after same has been dried it should be oiled. The indicator is a very delicate instrument, and upon its proper care depends its accuracy, hence its value, and too much attention cannot be bestowed upon its care and adjustment.

Testing the Instrument

Examine the instrument and try each part separately and see that it works smoothly. Put the instrument together without the spring. Hold the instrument by the steam cylinder in the right hand, and with thumb and finger raise the pencil lever very carefully to full extent of travel.

Place the thumb of right hand under the steam connection, release the pencil lever. Now slightly release the thumb over steam connection and note the fall of the piston. Repeat this until piston has traveled full stroke. The piston should fall freely every time the thumb is withdrawn. If, however, the piston moves in a sluggish manner, there is then excessive friction. If on the contrary it falls freely, we know that the friction is a minimum. Now withdraw the piston in the manner above described and put in the desired spring. Oil piston and connect up the instrument. Before placing instrument on cylinder or indicator cocks, blow out thoroughly the pipes and connections; too much care cannot be exercised in making sure that the connections are thoroughly cleansed, as any grit or dirt is not only liable to cut the cylinder, but it will affect the diagram as well.

Changing Indicator Springs

The remarks made under the head of care and adjustment explain the method sufficiently, and in this connection it is only necessary to add: Care must be taken to see that the spring is shouldered in cap, and full down on piston. In removing the spring on the outside spring instrument unscrew the knurled nut at the top until the end of the spring is released. Then turn the spring until it is free from the base. The piston is prevented from turning whilst removing the spring by inserting the pin (furnished with the instrument) in holes in the spring base.

The adjustment for atmospheric line when taking diagrams from condensing engine or low pressure cylinder of multiple expansion engines is made by the knurled nut at top.

Having described the instrument, its care and adjustment, we will now take up the connections to cylinders and reducing motions.

Cylinder Connections

Cylinders of marine engines are as a rule fitted with pipes and 3-way cocks.

The cylinders have bosses cast on them both top and bottom. The bosses are drilled through into the counter bore of the cylinder. The outer end is tapped for 1″ pipe; short nipples are screwed into the bosses, and ells used to connect with the side pipes. There is a great mistake in using ordinary ells, and wherever possible long-turn ells should be used, as the friction of steam is greatly reduced, and short bends should in all cases be eliminated.

The side pipes connect with a 3-way cock. Frequently angle valves are used in place of ells. This is very bad practice, and should not under any circumstance be countenanced.

When the pipes are to remain permanent fixtures, the 3-way cock is fitted with a screw cap, and when the instrument is not in commission, this cap should be screwed on to prevent any dirt, etc., getting into pipes.

The following should be remembered: Angle valves should never be used. The steam should be led to the instrument without any abrupt change of flow having to be encountered. In case the cylinder is not fitted with bosses, and holes have to be drilled in cylinder, the location of same must be such that the flow will not be disturbed, such as would occur by having holes opposite steam ports, as the inertia effect of the steam would affect diagram. Care must be exercised to see that cylinder head does not block the openings.

Where the stroke is very long, or pipes require a bend, short nipples with long-turn ells looking up should be used; the straight-way cocks

can then be screwed into these ells, and the instrument will then be in a vertical plane. Never use the instrument in a horizontal plane, that is to say, do not screw straight-way cock into the boss.

Never if possible use ordinary ells, use only long turn ells, and close nipple, and use two instruments to each cylinder. If the engine is to be indicated then the data should be accurate, and if it is not worth assuring oneself that every precaution has been taken to make it so, then do not attempt to reason about the diagrams taken.

Never use any lead or litharge in connecting the pipes, as it is liable to get into the steam cylinder of the instrument and ruin it. In making up the connections, use oil on pipe threads. If after assembling there is a leak, same can be eliminated by winding strands of waste around the exposed thread. The distortion of diagrams caused by long pipes is clearly shown in diagrams taken from George W. Clyde and the pipe arrangement before and after alteration is shown in figs. 1 and 2 of insert.

Reducing Motions

The reducing motion is as a rule, especially on the larger engines, a permanent fixture, and designed to give a length of diagram to suit the ideas of the designer. It should be designed to give a diagram not less than 4 inches long, except in high speed engines where the drum is a smaller diameter and hence a shorter diagram is a necessity.

The design of the motion is not a standard. Plate 1 shows the usual type of reducing motion. This is simply an arm or lever driven from the cross-head pin of the main engine through the medium of a short link. The lever is pivoted to the housings and pin for leading cord is located to give a certain length of diagram.

Another method of reducing the piston travel consists of a steel rod, pivoted to the cross-head pin; on the housing is bolted a bracket, to which is pivoted a brass sleeve; this sleeve carries an adjustable pin, to which the leading cord is attached by moving this pin in or out; the length of diagram can be varied. Still another method, and one which is in every way superior, is to drive a lever which is pivoted to either the housing or column, from the cross-head pin through the medium of a link. At the other end of the lever is connected a light vertical rod guided at its upper end by a guide bolted to the cylinder foot. This rod has on its upper end an eye into which the hook on the drum cord can be engaged or disengaged. This eliminates a long leading or driving cord, and the connection is therefore very short. This is an ideal motion, and as it can be made very light, and yet possess the requisite rigidity, the effect of inertia is too small to take account of.

Taking Diagrams

Before putting instrument on straight or 3-way cocks, blow out the pipes thoroughly, make sure there is no dirt or grit left in them. Remove the piston and parallel motion and connect the instrument to cock. See that leads are correct, and after adjusting same, screw the instrument down tight.

Adjust now the length of leading or driving cord, exercising care to see that drum does not hit the stops in either up or down stroke. After this adjustment has been made, see that the hook on the drum cord is secured without any danger of slipping. See further that the loop or ring on driving cord is secured against slipping. Open now the steam connection and blow steam through the cylinder. After having done this make sure no dirt is in the cylinder. Oil the piston with good cylinder oil as directed, and insert it in cylinder, screw down the cylinder cap. Turn steam on the instrument and let it work until all condensation is eliminated, and instrument is thoroughly warmed. When dry steam blows through the reliefs we are prepared to take diagrams; see that the joints in parallel motion are oiled with porpoise oil, as explained in previous pages.

Placing Cards on Drum

Take a blank card and turn over one end about ¼ inch. Insert this under one of the clips on drum, then with thumb and finger draw card around drum and place the other end of card in the second clip. With thumb and finger pull card down on drum until it touches the shoulder at base of drum, flatten both edges out by passing the finger down the turned edges, exercise care and see that card is tight and smooth.

After the adjustment of the pencil has been made and the drum put in motion, press the adjusting screw against stop, and describe the atmospheric line first. Pull pencil away from paper and then open cock to steam, press screw against stop, and do not permit pencil to travel more than once around the card. In other words, hold only for one revolution as near as can be judged. If 3-way cock is used, mark on card whether taken from top or bottom. If top, then repeat the process for bottom. After diagrams have been taken the data should be inserted in their respective places on back of diagram as shown in fig. 9. Pressing adjusting screw against stop is the same as saying pressing pencil against card, as it is supposed that the adjustment has been made as directed.

Before taking diagrams it is well to try the instrument to determine whether drum spindle is true. This can be done, as follows:

Place card on paper drum, press adjusting screw against stop and pull drum cord slowly by hand, describing the atmospheric line, return

drum to first position, open cock to steam, and with drum stationary describe the pressure line, with cock still open, again pull the paper drum, describing a line parallel with atmospheric line, with drum held in this position shut off steam, leaving the pencil to descend, open cock to atmosphere and we shall have described a rectangle. Now the admission line should be at right angles to the atmospheric line, and the steam line shall be parallel with atmospheric line. If the admission line is not at right angles with the atmospheric line, the drum spindle is not true. It is very important that this condition shall obtain. This test can be made before placing instrument on engine by removing the spring and raising the pencil lever by hand. The former

AMERICAN STEAM GAUGE AND VALVE MFG. CO.
BOSTON. NEW YORK. CHICAGO.
EXCLUSIVE MANUFACTURERS OF
American Thompson Improved Indicator.
(Original Thompson Indicator.)

December 1st 1908

DIAGRAM from M S. S. Admiral Engine 18"-28"-45"

Diameter of Cylinder 18"	Built by Wm Jones
Length of stroke 30"	Boiler Pressure designed 160
Revolutions per Minute 125	Barometer Reads 14.7 inches
Pressure of Steam in lbs. in Boiler 160	Throttle
Position of Throttle Valve wide Open	Regulator
Vacuum per Gauge in inches 26	REMARKS: Drop between Boiler
Temperature of Hot Well 120°	and H.P. Piston 15 lbs.
Scale of Spring 80	Wire drawing Excessive
Inside Diameter of Feed Pipe 6"	H.P. Piston Valve leaks.
" " Exhaust Pipe 8"	
Piston Valve on H.P. Cyl	

Fig. 9

method is to be preferred as the instrument has been warmed and everything in condition. If a test gauge can be attached at a point close to instrument, we can then determine whether our springs are correct. It is a good method to make this test before taking diagrams, and keeping the test card with other records.

Before proceeding to take up the subject of indicator diagrams, it will be well to give a description of the planimeter and its use.

Planimeter

The planimeter as its name implies is an instrument for the measuring the areas of irregular figures. There are several different types of instruments manufactured. We will, however, confine ourselves to the Amsler instrument as manufactured by the American Steam Gauge and Valve Manufacturing Company (see Fig. 10). This instrument consists of three essential parts, namely: A guide arm pivoted at "A" to the paper; a tracing arm which is hinged to the guiding-arm, and which carries the tracing point "B"; a measuring wheel "G," which carries a graduated cylindrical scale. There is also a vernier "E" for reading the scale on the wheel.

When in use the planimeter rests on the paper at three points. The pivot "A" which is a needle point pressed slightly into the paper; the edge of the measuring wheel "G," and the tracing point "B." A weight over the pivot "A" holds the needle point down, and gives the instrument stability.

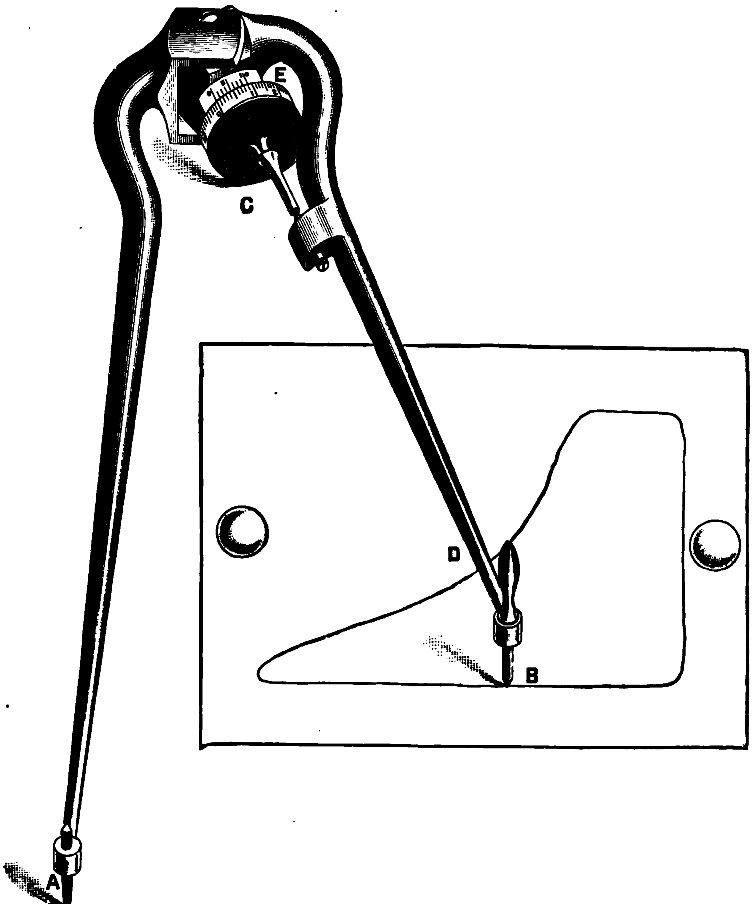


Fig. 10

To measure the area of any irregular figure like an indicator diagram the instrument is placed as in Fig. 10, so that the arm shall not take inconvenient positions when the outline of the diagram is traced. Take any point on the diagram as at "B" and set the measuring wheel to read zero, trace the diagram in a clockwise or right-hand direction.

Before proceeding to explain the method of reading, it will be as well to describe the vernier and measuring wheel.

Let Fig. 11 represent a scale of units numbered 1, 2, 3, 4, etc., which

are sub-divided into tenths. The vernier U. V. is as long as nine of the sub-divisions, and is divided into ten parts. Thus the intervals of the vernier are 9/10ths as long as the interval of the scale, or we can say they are 1/10th of an interval shorter. As shown the index of the vernier reads 4.5 on the scale. It will be noted that the 4th division of the vernier coincides with a division of the scale, the 3d division of the vernier is 1/10th of an interval from the next mark on the scale, the 2nd division is 2/10ths, etc. Therefore, the reading of the vernier is 4.54 square inches, for if the measuring wheel is divided into ten equal parts, each to equal one square inch, then the sub-divisions enable us to read to hundredths of a square inch.

Therefore, starting at any desired point run tracing point "B" in clockwise direction, and trace around diagram until starting point

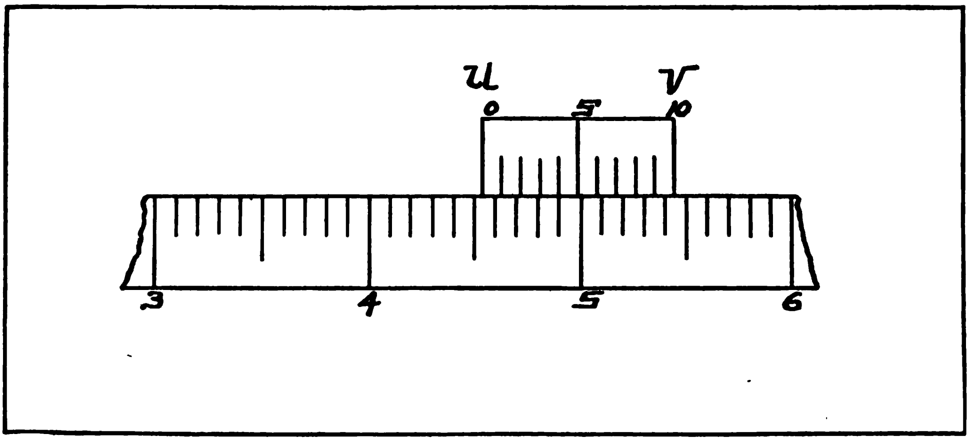


Fig. 11

is reached, find highest figure on measuring wheel which has passed the zero on vernier moving to the left, in this case 4. Find next the number of completed divisions between 4 on measuring wheel and zero on vernier, which is in this case 5. Find division on vernier which corresponds with same division on measuring wheel, and in this case it is 4. Therefore, the exact reading is 4.54 square inches.

After the operator becomes familiar with the instrument it is not necessary to set the wheel to zero, but take the reading before starting to trace outline of diagram, and subtract this from the final reading. Thus, suppose when instrument is in position we find the reading to be 1.64, the final reading is 6.18. Therefore, 6.18 − 1.64 = 4.54 square inches, area of card.

The instrument can be used for finding areas of any irregular figures. If the area is large, divide it by lines into areas of less than 20 square

inches and take separate measurements. If drawing be to scale multiply the reading of instrument by the square of the ratio number of the scale. Should it be required to find the area of an irregular figure containing 6 square inches drawn to a scale of 3 inches = 1 foot 3 inches = 1 foot is $\frac{1}{4}$ size. Therefore, $\frac{1}{4} \times \frac{1}{4} = \frac{1}{16}$ and $6 \times 16 = 96$ square inches.

Definitions

Relating to indicator diagrams. (See Fig. 12.) Four phases of valve-motion occur during a complete revolution of the engine, and are as follows:

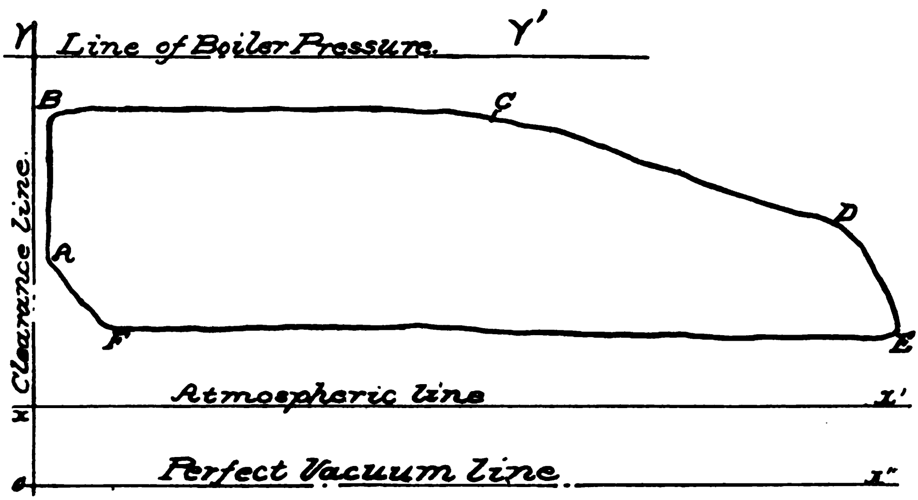


Fig. 12

Admission ABC. When valve is open, and steam passing into the cylinder.

Expansion CD. When valve has cut off the steam supply to cylinder, and hence steam is neither admitted or released, therefore, the piston is moved through this distance by the expansive force of the steam.

Exhaust DEF. When the valve closes the admission port, and the port to exhaust opened, and hence steam is escaping from cylinder into receiver, or condenser if condensing, or atmosphere if non-condensing.

Compression FA. When all ports are closed, and the remaining steam in the cylinder acts as a cushion to bring the piston gently to rest.

The atmospheric line XX′ is a line drawn by the pencil of the indicator when both sides of the piston are open to the atmosphere. The steam is of course shut off from instrument. The atmospheric line on the diagram represents the pressure of the atmosphere, the gauge reading being zero.

The vacuum line OX″ is a reference line drawn at a distance corre-

sponding to barometer-pressure by scale below the atmospheric line. The barometric pressure which is usually 14.7 lbs. This line represents a perfect vacuum, or absence of pressure when drawn to scale to 15 lbs.

The clearance line OY is a reference line drawn at a distance from the end of the diagram equal to the same per cent. of its length as the clearance or volume not swept through by the piston is of the piston displacement. In other words, the distance between the clearance line and the end of diagram represents the volume of the clearance between piston and cylinder head, plus the volume of ports and passages at that end of cylinder.

Line of boiler pressure YY′ is a line drawn parallel to the atmospheric line, at a distance from it by scale equal to the boiler pressure shown by gauge.

Admission line AB is the line showing the rise of pressure due to admission of steam to the cylinder by the opening of steam valve.

Point of admission A indicates the pressure when the admission of steam begins at the opening of the valve.

Steam line BC is drawn when the steam valve is open and steam is being admitted to the cylinder.

Point of cut-off C is the point where the admission of steam is stopped by the closing of the valve.

Expansion curve CD shows the fall in pressure as the steam in the cylinder expands.

Point of release D shows where the exhaust valve opens.

Exhaust line DE shows the change in pressure which takes place when the exhaust valve opens.

Back pressure line EF shows the pressure acting against piston during its return stroke.

Point of exhaust closure F is the point where the exhaust valve closes. Point of compression F is where the exhaust valve closes, and compression begins. Compression curve FA shows the rise in pressure due to compression of the steam remaining in the cylinder after the exhaust valve has closed.

Initial pressure is the pressure acting on the piston at the beginning of the stroke.

Terminal pressure is the pressure above the line of perfect vacuum which would exist at the end of the stroke if the steam had not previously been released.

Admission pressure is the pressure acting on the piston at end of compression, and is as a rule less than the initial pressure.

Compression pressure is the pressure acting on the piston at beginning of compression; it is the least back pressure.

Cut-off pressure is the pressure acting on the piston at beginning of expansion.

Release pressure is the pressure acting on the piston at end of expansion.

Mean forward pressure is the average height of that part of the diagram traced on forward stroke.

Mean back pressure is the average height of that part of the diagram traced on the return stroke.

Mean effective pressure is the difference between the mean forward pressure and the mean back pressure during a forward and return stroke.

It is the height or length of the mean ordinate intercepted between the top and bottom lines of the diagram multiplied by the scale of spring used in instrument when diagram was taken. It is obtained without regard to atmospheric or vacuum lines.

Equivalent or referred mean effective pressure, often written as aggregate equivalent pressure referred to low-pressure cylinder, is the mean effective pressure which would be required to produce the same indicated horse-power from a cylinder of the same dimensions as the low-pressure cylinder of a multiple expansion engine.

Ratio of expansion is the ratio of the volume of steam in the cylinder at the end of stroke to that at cut-off.

Initial expansion is the fall of pressure during admission due to imperfect steam supply.

Wire drawing is the fall of pressure between admission and cut-off.

Horse-power. The unit employed to measure the rate at which work is done in a steam engine is the "horse-power," the power exerted in the performance of 33,000 foot pounds of work per minute.

A distinction must be made between the indicated horse-power and the actual or brake horse-power. When we speak of indicated horse-power, the work done per minute by the steam on the piston of the engine, as computed from indicator diagrams, is understood. The friction of the shafting and pumps, as well as the reciprocating parts, friction of piston rods through stuffing boxes, glands, etc., valve gear and all working parts, absorb power and cause a loss which is termed frictional losses.

If, therefore, the sum of all these frictional losses is deducted from the indicated power, we get the actual power available, which is delivered to the screw propeller, or in other words it is the rate at which useful work is done in turning the propeller.

The brake horse-power in very large engines is less, and in small engines considerably less than the indicated horse-power.

Now, brake horse-power $\div$ indicated horse-power = efficiency of engine. Therefore, efficiency of engine multiplied by indicated horse-power = brake horse-power. Stated in form of an equation we have: B. H. P. = N $\times$ I. H. P. when N = efficiency.

The following table (calculated from Middendorf, Scheffswiderstand und Maschinenleistung) gives values of efficiency N:

I. H. P.	N	I. H. P.	N
5 to 10	0.58	600 to 700	0.71
10 to 50	0.59	700 to 800	0.72
50 to 100	0.60	800 to 900	0.73
100 to 150	0.61	900 to 1,000	0.74
150 to 200	0.62	1,000 to 2,000	0.79
200 to 300	0.64	2,000 to 3,000	0.85
300 to 400	0.66	3,000 to 4,000	0.88
400 to 500	0.68	4,000 to 5,000	0.90
500 to 600	0.69	6,000 and over	0.91

The determining of the brake horse-power has been, until recently, a difficult and in fact almost impossible procedure due to the fact that large powers had to be absorbed, and the difficulties of fitting a brake to absorb it very great. The values of the efficiency as shown above have been taken as approximate values, and until recently approximate values were the only ones available.

The torsion meter enables us to determine accurately the power delivered to the shaft. The latest trials made with the torsion meter have given the following values:

I.H.P.	N	I.H.P.	H.
1,630	0.885	2,370	0.920
1,640	0.091	2,690	0.911
1,940	0.911	4,500	0.935

Before entering upon the subject of the indicator diagram, it will be as well if we explain the rules of mean ordinates.

The simplest way of determining the M. E. P. is by the planimeter. It frequently happens that we are compelled to compute the pressure without the assistance of this instrument, hence we have to resort to some practical method of computation.

"Rule of Mean Ordinates"

Divide the diagram into ten equal parts by lines at right angles to the atmospheric line, and measure the center of each division between the top and bottom lines forming the diagram. The mean height of the ten divisions, measured in inches and multiplied by scale of spring, is equal to the mean effective pressure in pounds per square inch. Greater accuracy is obtained by dividing diagram into 20 equal parts and measuring each ordinate, dividing the sum by 20 to obtain mean ordinate, then multiply by scale of spring. In the use of the planimeter we get the area of the diagram, and dividing it by the length of card we get the height of the mean ordinate, and multiplying this mean

ordinate by scale of spring as explained gives us the M. E. P. in pounds per square inch.

Fig. 13 shows the method of obtaining the M. E. P. and dividing the card.

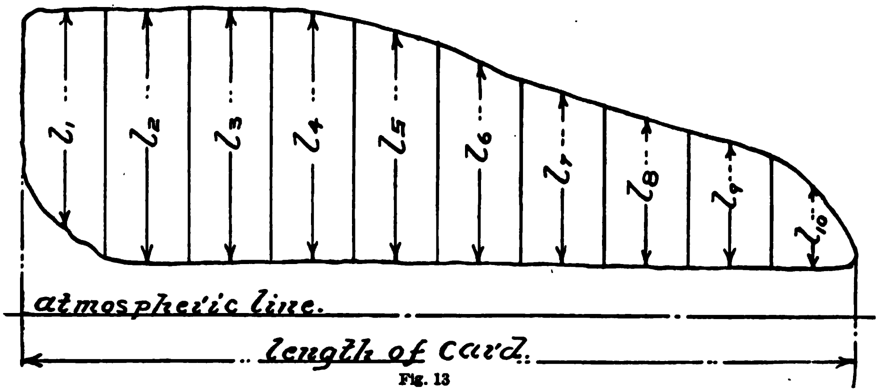


Fig. 13

Numb. of Ord.	Length of Ord.
L_1	1.09375″
L_2	1.3125 ″
L_3	1.3125 ″
L_4	1.3125 ″
L_5	1.1875 ″
L_6	1.0625 ″
L_7	.90625″
L_8	.40625″
L_9	.65625″
L_{10}	.4375 ″
	Sum = 9.68750

Lgt. of Mean Ord. = 10)9.68750(= 0.96875
Scale of Spring = 60 lbs. per inch.
Mean Effective Pressure = $0.96875 \times 60 = 58.125$ lbs.
Mean Effective Pressure by Planimeter = 58.37 lbs.

Simpson's Rule

Another method is by what is known as Simpson's Rule, and is as follows:

Divide the diagram into ten equal parts as before, and lettering the ordinate as shown, and take,

$Y_0 + Y_{10} = L_1$
$Y_1 + Y_3 + Y_5 + Y_7 + Y_9 = L_2$
$Y_2 + Y_4 + Y_6 + Y_8 = L_3.$

The mean effective pressure in pounds per square inch will therefore be,

$$\frac{L_1+4L_2+2L_3}{30}\times S$$

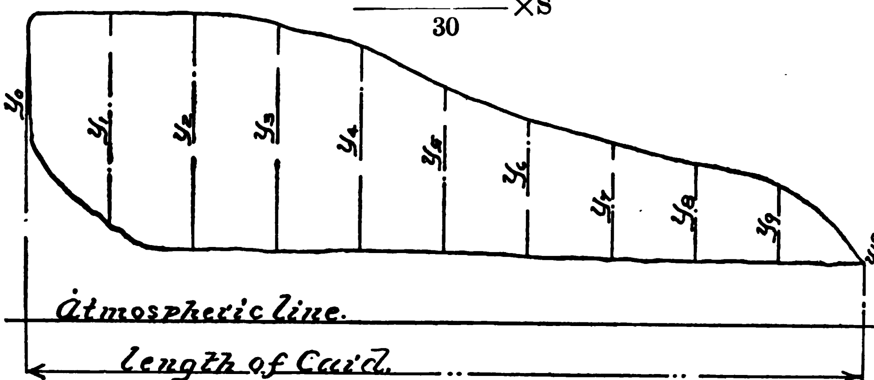


Fig. 14

Simpson's first rule is: To the sum of the first and last ordinate, add four times the even ordinates, plus twice the odd ordinates and multiply the sum by one-third the common interval gives area of figure. Now our interval is one-tenth, and one-third multiplied by one-tenth is equal to one-thirtieth, and this one-thirtieth multiplied by the scale of spring gives the divisor of our fraction. Therefore, the sum of $L_1+4L_2+2L_3$ divided by one-thirtieth multiplied by spring gives the mean effective pressure in pounds per square inch. Computation in full of Fig. 14.

Numb. of Ord.	Length of Ord.	Multiplier	Function of Ord's.
y_0	0.25″	1	0.25
y_1	1.125″	4	4.5
y_2	1.218″	2	2.436
y_3	1.1875″	4	4.75
y_4	1.0625″	2	2.125
y_5	.875″	4	3.5
y_6	.71875″	2	1.4365
y_7	.625″	4	2.5
y_8	.5″	2	1.
y_9	.375″	4	1.5
y_{10}	0.0	1	0.0
		Sum of function,	23.9975

Common interval $=\frac{1}{10}$

$\frac{1}{3}$ " " $=\frac{1}{3}\times\frac{1}{10}=\frac{1}{30}$.

$23.9975\times\frac{1}{30}=30\ )23.9975(\ =0.7999$.

Scale of Spring = 60 lbs. per inch.

Mean Effective Pressure = 0.7999 × 60 = 47.994 lbs.

Mean Effective Pressure by Planimeter = 48.7 lbs.

Engine Types

Single-cylinder engines are those in which the whole work of the steam is performed in one cylinder. Twin-cylinder engines are those in which each cylinder works in precisely the same way as a single-cylinder engine; the steam passing into both cylinders direct from the boilers, and exhausting from both cylinders into the atmosphere or condenser.

Compound engines are those in which the steam works successively in two or more cylinders placed close to each other.

In a two-cylinder compound engine the steam passes from the boiler into the high-pressure cylinder, exhausting from the high-pressure cylinder into the receiver and thence into the low-pressure cylinder. From the low-pressure cylinder it exhausts into the condenser.

In a triple expansion engine, the steam passes from the boiler into the high-pressure cylinder, exhausts from the high-pressure into the first receiver, from thence into the intermediate cylinder, exhausting from the intermediate cylinder into the second receiver, from thence into the low-pressure cylinder, and from low-pressure cylinder into the condenser.

In a quadruple expansion engine the steam passes from the boiler into the high-pressure cylinder, exhausts from high-pressure into the first receiver, from thence into the first intermediate cylinder, exhausts from first intermediate cylinder into the receiver and from there into a second larger intermediate cylinder, exhausting from the second intermediate cylinder into the receiver, thence into the low-pressure cylinder, and from the low-pressure cylinder into the condenser.

As the steam decreases in pressure in passing through the various cylinders, its volume correspondingly increases; therefore the cylinder, from high-pressure onward, must increase in size, this increase depending upon the degree of expansion.

It frequently happens that the same degree of expansion may be divided between two cylinders, either two high-pressure or two low-pressure cylinders. This is resorted to for constructive reasons.

A triple expansion engine may have four cylinders high-pressure, intermediate-pressure, and two low-pressure cylinders of the same size.

A triple expansion engine having 5 cylinders, namely, two high-pressure, one intermediate, and two low-pressure cylinders, has been installed in large Atlantic liners.

Multiple expansion engines are computed in precisely the same manner as a single cylinder engine. The reasoning is the same as if all work of the steam were done in the low-pressure cylinder. This will be more readily understood when we take up the computations of Equivalent M. E. P. and Cylinder Dimensions.

CHAPTER II

Work of Steam

It is necessary that the work of the steam in the cylinder is comprehended thoroughly, and it will therefore be necessary to consider a hypothetical case. Let us assume that we have a vertical cylinder, open at the upper end to the atmosphere, and closed at the bottom. We will further assume that the cylinder is fitted with a piston without weight and frictionless.

If a certain quantity of water is introduced at the bottom of the cylinder and a fire is built under it to convert the water into steam, we will have the boiler and engine represented by one vessel, the piston and water being brought into direct contact.

Let us make the diameter of piston about 13½ inches; this will give us a sectional area of 1 square foot, equal to 144 square inches.

Let a quantity of water weighing 1 pound be poured into the cylinder, and let this stratum of water support the piston.

As the upper end of the cylinder is open to the atmosphere, the pressure of the atmosphere (here taken as 14.7 lbs.) acts upon the piston, amounting to 14.7 lbs.$\times$144 square inches$=$2,116.8 lbs. on the square foot of surface of the piston. The temperature of the water under atmospheric pressure will be raised to 212° F, before any steam is generated. If now the heat of the fire be maintained, the temperature will remain stationary at 212° F, but steam will be formed, and disengaged under the piston. The piston supposed to be frictionless and without weight will be raised with its load of 2,116.8 pounds through consecutive stages, each, say, one foot, until it reaches an elevation of 26.6 feet above the bottom of the cylinder. When this point is reached we shall have found the whole one pound of water evaporated, the constant elasticity of the fluid having been measured by 14.7 pounds per square inch, and a temperature of 212° F.

What are we to understand by this? We see that the pound of water has been entirely evaporated into steam of atmospheric pressure, and occupies a volume of 26.6 cubic feet, for 1 square foot area$\times$26.6 feet$=$26.6 cubic feet. The initial work consists in having lifted a weight of 2,116.8 pounds through a height of 26.6 feet, or, expressed in foot pounds, 2,116.8 pounds$\times$26.6 feet$=$56,306.88 foot pounds.

The above demonstration affords a vivid conception of the expansive force of steam, or to be more exact, the force of water when converted into steam. Here we had a lamina of water not quite one-fifth of an inch in depth, lying at the bottom of a cylinder 13½ inches

diameter. This water is converted into steam of atmospheric pressure of 1,602.4 times its original volume, for $\frac{1}{5}$inch = 0.0166 feet, and 26.6 feet ÷ 0.0166 feet = 1,602.4.

As one heat unit is equivalent to 778 foot pounds, the value of the external work expressed in heat units is 56,306.88 foot pounds ÷ 778 heat units = 72.37 H. U. There is a small expenditure of energy in raising the mass of steam against the force of gravity. Thus, the average height to which the steam is raised is 26.6 ÷ 2 = 13.3 feet, and 1 pound × 13.3 feet = 13.3 foot pounds, or, 13.3 foot pounds ÷ 778 H. U. = 0.017 H. U.

British Thermal Unit

A British thermal unit or B. T. U. is the heat required to raise one pound of water from 62° F. to 63° F. Heat is always measured in B. T. U.'s in the English system.

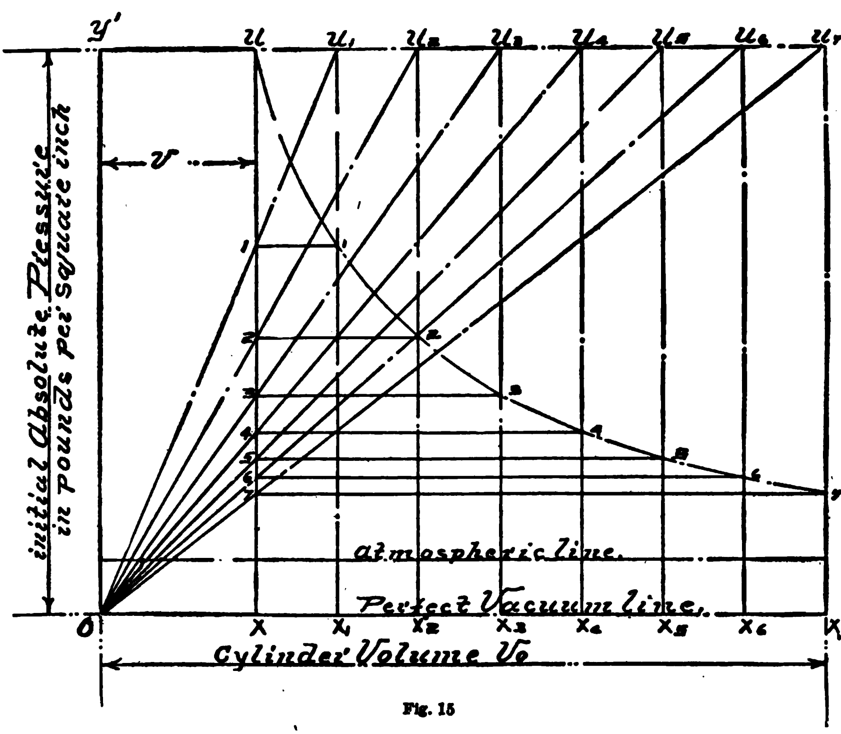


Fig. 15

Expansion of Steam

The steam in the cylinder of a steam engine during expansion is supposed to follow substantially a law known as the law of Boyle and Mariotte. This law states that the pressure varies as the volume in an

inverse ratio. That is to say: As the volume increases the pressure suffers a decrease.

Symbolically, if P = pressure, and V = volume, then P. V. = C.

We say substantially, because the actual changes of pressure do not follow the law exactly. The pressure may, and in the majority of cases it does fall more rapidly in the early stages of the expansion, and less rapidly in the latter portion than indicated by the law of inverse ratio. Therefore, the finial pressure is as a rule greater than that which would be deduced from the ratio of expansion.

Now the fullness of the expansion curve depicted on the indicator diagram, near the end, compensates for the hollowness near the beginning, and hence we find that the area bounded by the curve is practically equal to that bounded by a hyperbolic curve according to the law.

We, therefore, assume that for all practical purposes, and for general investigation, the steam expands according to the law, P. V. = C.

The curve which represents diminishing pressures due to increasing volume is a portion of a hyperbola.

The rectangular hyperbola used as a curve of expansion is constructed as follows: (See Fig. 15.)

Let $OY' = P$, the initial pressure.

Let $Y'U = V$, the volume up to cut-off.

Let $OX_7 = Vo$, the volume at end of stroke.

Produce the line $Y'U$ to U_7; divide UU_7 into any number of parts, say 7. Draw a series of radiating lines from O to U_1, U_2, U_3 ...... U_7.

Now where the radiating lines OU_1, OU_2 ...... OU_7 intersect the ordinate UX, such as points 1, 2, 3, etc., these points of intersection give points through which are drawn lines parallel to OX_7, as 1, 1,–2, 2, – 3, 3, etc.

Drawing a fair curve through the corresponding points of intersection with the ordinates U_1 X_1, U_2 X_2, U_3 X_3 ...... U_7 X_7, we have the curve known as the rectangular hyperbola, or curve of P. V. = C.

To determine the pressure at any point of the expansion curve, say for volume $Y'U_3 = OX_3$. Draw the diagonal line OU_3, then through point 3 the intersection of U, X and OU_3 draw the horizontal line 3,3 parallel to OX_7. Point 3 is a point on the expansion curve and the vertical line 3, X_3 gives the absolute pressure corresponding to the volume OX_3.

Should we desire to obtain the finial pressure after expansion: Draw the diagonal line OU_7; then through the point 7, the intersection of UX and OU_7, draw the horizontal line 7, 7, parallel to OX_7. The vertical line 7, X_7 gives the required finial absolute pressure. We can conversely find the volume which a quantity of steam V. would

occupy at the pressure P. if it were compressed to the pressure P_a. To obtain the volume, draw the diagonal line OU′ (see Fig. 16) now where OU′ intersects Y′U, draw A, A parallel to Y″O. The line Y″A gives the required volume.

It should be borne in mind that Y′U is volume without clearance, and OX_7 is vacuum line.

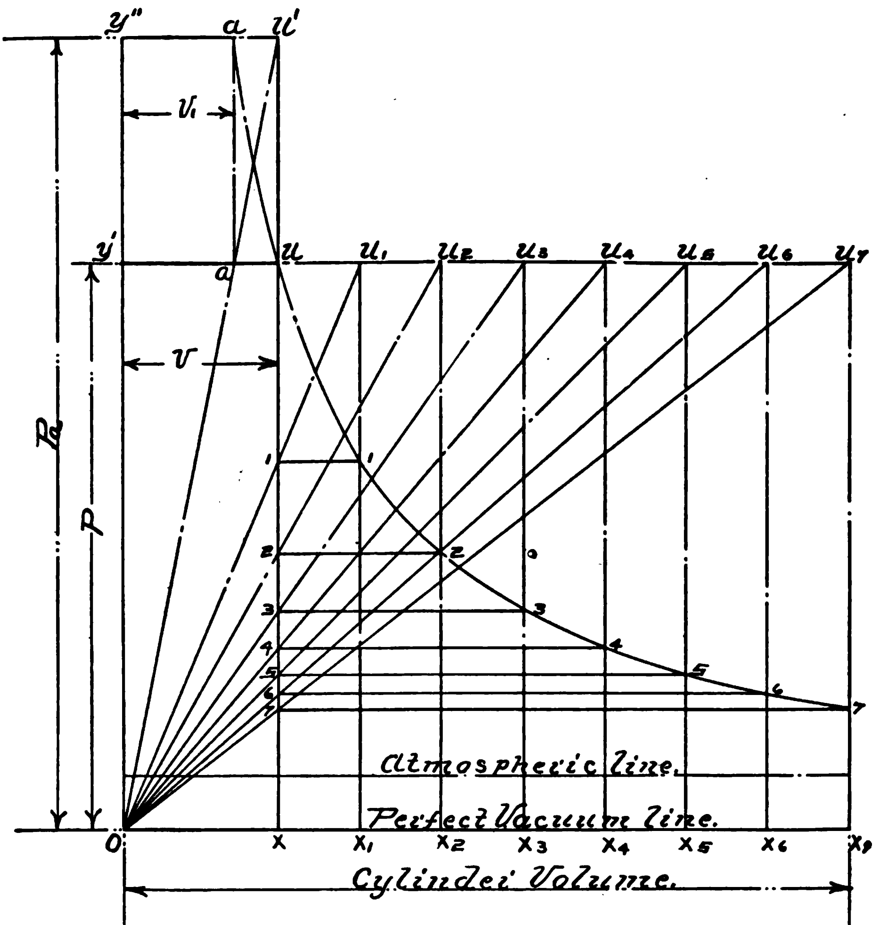


Fig. 16

To illustrate the application of the hyperbolic law of expansion, showing that the product of pressure and volume at any point of the expansion-curve is constant. Let the line XX1 (Fig. 17) represent the stroke of the piston and the corresponding volume described by it without clearance.

Assume steam of 160 pounds absolute pressure be admitted for a space 1 foot in length XA. The area of the rectangle is the product of the pressure and volume of the steam admitted. If the steam expands to double its volume XD the pressure will be one half, represented

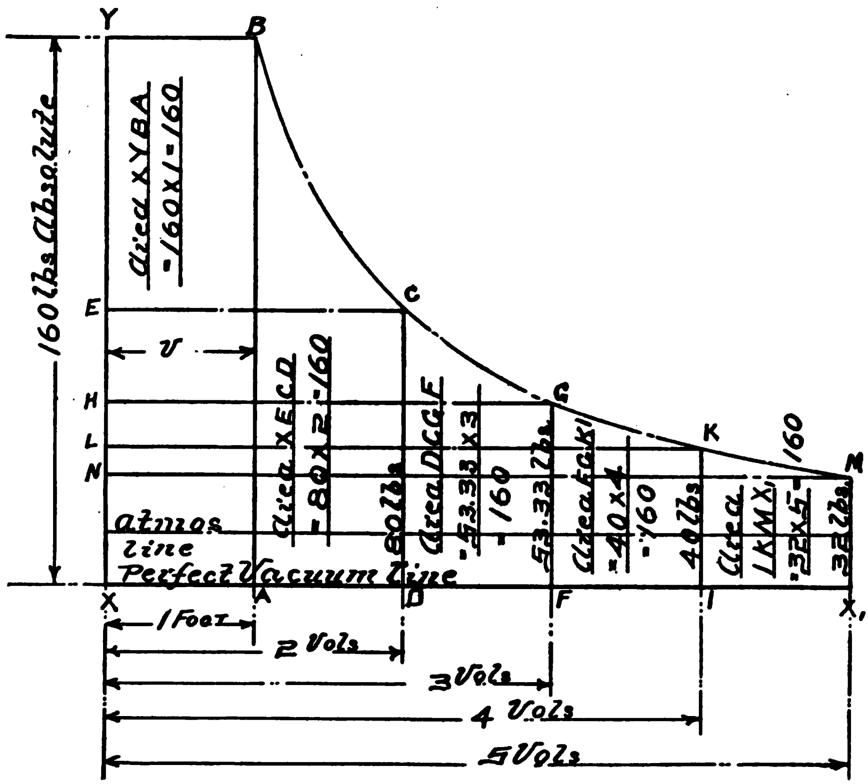


Fig. 17

by DC. The area of the rectangle XE×XD, is the product of pressure × volume, and this area will be equal to the area of the rectangle XY×XA.

Expanding further to any number of volumes we find the pressure multiplied by volume is equal to the initial pressure multiplied by initial volume. The area of each rectangle is therefore equal to the original rectangle. The hyperbolic curve containing these rectangles may be indefinitely extended at either end, embracing toward the left hand, high pressures and small volumes, and to the right hand, low pressures and large volume.

The area of the rectangle XYBA, being the product of pressure and volume, expresses the work done upon the piston by the steam on

entering the cylinder and occupying a given volume. The area bounded by the hyberbolic curve BM, the ordinates MX_1, AB, and the base AX_1 expresses the work done by expansion of the steam after the communication with the steam supply has been cut off.

Let P = absolute initial pressure of steam.

Let V = volume up to cut-off.

The work done by the steam during admission is P.V. (See Fig. 18.)

Let S = whole stroke.

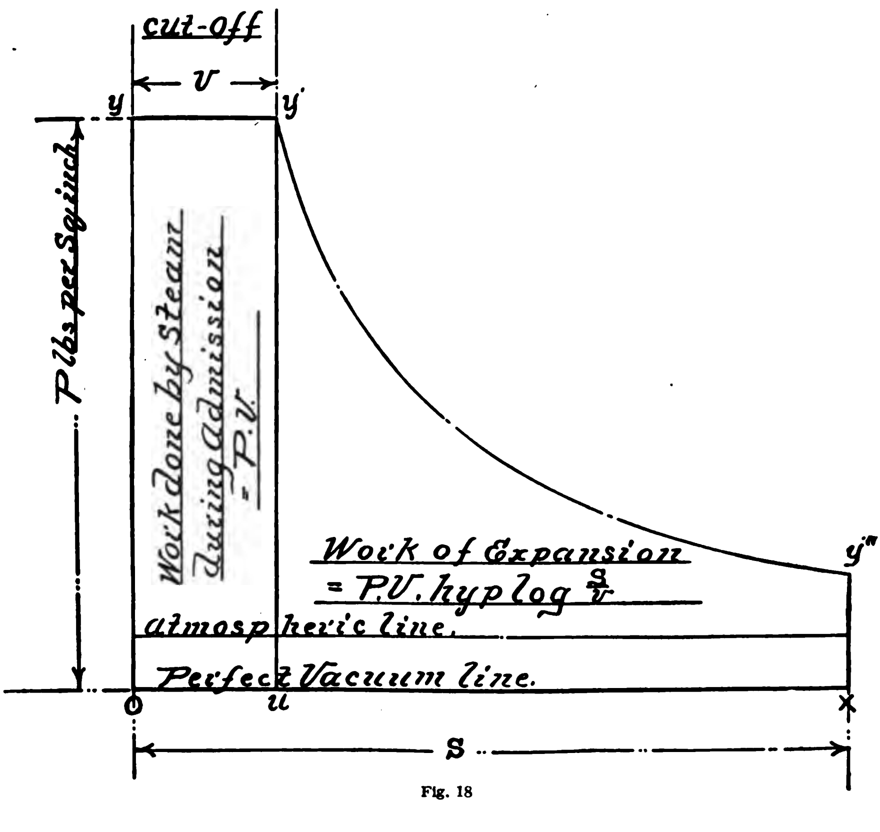


Fig. 18

The mean pressure during this period, in relation to the whole stroke S, is $p = P\frac{V}{S}$ where p = mean pressure.

The work of expansion is equal to the area Y′Y″XUY′. The area Y′Y″XUY′ = P V hyp log $\frac{S}{V}$. The mean pressure during the work of expansion in relation to whole stroke S is P $\frac{V}{S}$ hyp log $\frac{S}{V}$. Now $\frac{V}{S}$ = cut-off = C.

C is expressed either as a fraction or as a percentage of the volume of the cylinder. Thus, cut-off ¼ stroke = 4)1.00(= 0.25 or 25 per cent.

of stroke. $\frac{S}{V}$ is termed the ratio or degree of expansion. The ratio or degree of expansion is also equal to $\frac{1}{C}$ or 1 divided by the cut-off.

It should be clearly understood that in multiple expansion engines, that is, compound, triple and quadruple expansion engines, the term total cut-off is frequently used, and is understood to mean the ratio that the volume of steam admitted to the high-pressure cylinder bears to the volume of the low-pressure cylinder.

Total expansion means the ratio that the volume of the low-pressure cylinder bears to the volume of steam admitted to the high-pressure cylinder.*

As an example, suppose we have a triple expansion engine, the volume of the low-pressure cylinder is 7 times the volume of the high-pressure cylinder. The ratio of cylinder capacities are therefore 1:7.

Assume a cut-off in high-pressure cylinder of 75 per cent. of stroke. The ratio or degree of expansion is $\frac{7}{0.75}=75)700(9.33$.

And the total cut-off will be $\frac{1}{9.33}=\frac{.75}{7}=0.107$.

The cut-off in the high-pressure cylinder is equal to the ratio of cylinder capacities ÷ total expansion.

Thus $\frac{7}{9.33}=0.75$.

Let C = total cut-off.

Let C_h = cut-off in the high-pressure cylinder.

Let R = ratio of the volume of low-pressure cylinder to that of the high pressure cylinder.

Then total cut-off $C=\frac{C_h}{R}$.

And total expansion $=\frac{1}{C}=R\frac{1}{C_h}$.

Clearance

All engines have clearance, the space between the piston and cylinder-head when piston is at either end of its stroke. The steam passages between valve face and cylinder bore. This clearance space must be filled with steam of the initial pressure at the beginning of each stroke. The clearance is measured as a certain percentage of the cylinder volume. When so expressed it is termed volumetric clearance. For example, if we have a cylinder 12 inches in diameter by 12 inches stroke: The volume of the cylinder = area of cylinder in square inches × stroke in inches. Now the area of a 12″ circle = 113.10 square inches. 113.1□″ × 12″ = 1357.2 cubic inches volume of cylinder.

Suppose the clearance between cylinder head and piston plus the clearance in port is equal to 95 cubic inches. The percentage is, therefore, 95 ÷ 1357.2 = 0.07 or 7 per cent. It is rather a tedious and sometimes impossible task to determine accurately the correct clearance, and

* The volume of a cylinder is equal to the area of the cylinder in square inches multiplied by the stroke of piston in inches.

where the data must be very accurate, the only way to determine it is from the cylinder drawings. The clearance may be measured in parts of the stroke and the clearance length added to the period of admission. It is evident that this sum represents or expresses the initial volume of steam for expansion.

Thus suppose that the clearance is 7 per cent. of the volume of the cylinder or piston displacement, which is one and the same thing, and let us further assume cut-off at half stroke = 50 per cent.

We readily see that the effective cut-off is not 50 per cent., but it is more than this by the amount of clearance, and hence we have the

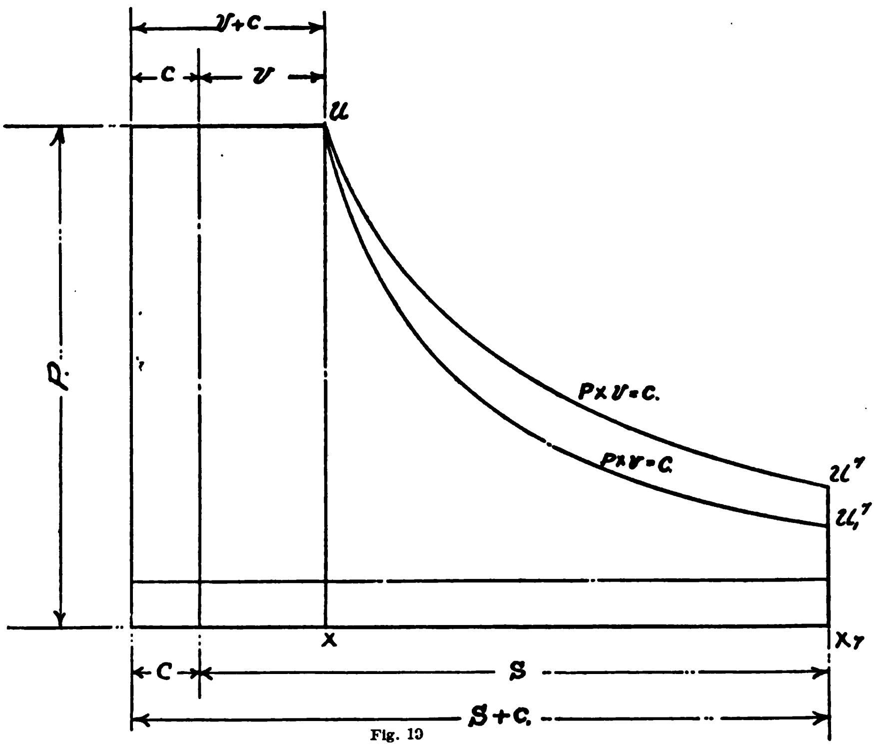


Fig. 10

expansion of a volume of steam equal to 50 per cent. plus 7 per cent. = 57 per cent. instead of only 50 per cent. This practially amounts to making the cylinder 7 per cent. longer and cutting off at 50 per cent. of the stroke without clearance.

The mean pressures in practice are greatly effected by clearance. Before the incoming steam can force the piston out, it has to fill the clearance space. Now this space being filled alternately with admission steam of a high temperature, and the cooler exhaust steam

having a lower temperature causes considerable loss by condensation during admission. It matters not how accurately the engine is designed, the clearance spaces are large, and the superficial areas, exposed to extreme variations of temperature, are likewise large. It will therefore readily be seen that clearance affects expansion prejudicially, due to the fact that it raises the terminal pressure, and affects compression, because it reduces the finial pressure of compression.

Diagram (Fig. 19) shows first the work of expansion is increased by clearance. Thus area XUU^7X_7X is greater than $XUU_1{}^7\ X_7X$, area $XUU_1{}^7\ X_7X$ representing work done during expansion without clearance. "Second," showing that compression must be increased to obtain a given terminal pressure if there is clearance.

The rate of expansion taken without clearance is termed nominal rate of expansion.

The rate of expansion taken with clearance is termed the actual rate of expansion.

When the clearance can be accurately determined it is better to use it, and obtain the actual instead of the nominal rate of expansion.

Then if V_n = nominal rate of expansion.

V_a = actual rate of expansion.

C = clearance as a fraction of the cylinder capacity.

We have $\frac{1}{V_a} = \dfrac{\frac{1}{V_n} + C}{1 + C}$. $V_a = V_n \dfrac{1 + C}{1 + CV_n}$.

$\frac{1}{V_n} + C$ is the volume of steam at cut-off between the piston and valve.

This steam expands to the volume $1 + C$ at the end of the stroke. If there is no compression of the steam before admission, the whole space $\frac{1}{V_n} + C$ must be filled with fresh steam at each stroke.

In some cases there is sufficient compression to fill the clearance space with steam of initial pressure. The volume of steam used during each stroke will then be that swept by the piston up to cut-off only. This will then be equal to $\frac{1}{V_n}$.

Whilst clearance serves to increase the mean pressure beyond that due to the nominal rate of expansion, it cannot be considered as a source of loss, unless the equivalent cut-off is taken to obtain the rate of expansion. With the use of higher steam pressures and higher rates of expansion the disadvantageous influence of clearance is diminished.

With good steam distribution and proper compression, the drawbacks due to clearance may be lessened. As the actual total cut-off deviates less from the theoretical, the limit of total expansion due to clearance can be arranged to fall in more favorable position. The clearance should, however, in any case be made as small as possible.

Losses in Cylinders

The principal causes of loss of pressure in the cylinders of a marine engine are the following:

Friction in boiler stop valve.
Friction in throttle valve on cylinder.
Losses by friction in main steam pipe.
Friction or wire drawing of the steam during admission.
Liquification during expansion.
Compression and back-pressure.
Friction in the ports and pipes.

The loss by friction in the stop-valves, throttle-valve, and main steam pipe does not show on the indicator diagram, but the loss is manifest in the fall of pressure or drop between boiler and piston.

The loss by friction or wire drawing is as a rule due to defective design and adjustment. Defective design embracing small steam ports. Valve chest too small, causing thereby expansion of steam into cylinder when valve opens without being replaced with sufficient rapidity by steam from boiler.

Adjustment embracing valves, not permitting a sufficiently large opening for the quantity of steam required. Valves not cutting-off with quickness. This latter is a defect inherent in a link motion.

Liquification during expansion, due in part to the cooling action of the cylinder walls.

In multiple expansion engines liquification losses are less than in single-cylinder engines. Exhausting before the piston reaches end of its stroke, whilst conducive to good working of fast-running engines, nevertheless shows a loss in the indicator diagram.

The Steam Jacket

The steam jacket is seldom used except for warming the engine cylinders. The value of the steam jacket decreases with the diameter of the cylinder and high piston speeds. The wet steam supplied by the average water tube boiler neutralizes the good effects.

Again it is only the innermost layers of the cylinder walls that are affected by the fluctuation of temperature taking place in the cylinder. The variations will be less in the outer layers of metal; each concentric layer has a mean temperature, diminishing toward the exterior surface of the walls. It is readily seen that the outer layers approximate to the surrounding temperature of the atmosphere. The higher the temperature the less far will the variations of temperature extend outward through the walls, and hence the exchange of heat during one revolution will be smaller.

Effective Mean Pressure With Clearness

Assume steam pressure = 100 pounds gauge or 100+15 = 115 pounds absolute.

Let clearance space equal one-ninth of the cylinder volume.

Back-pressure assumed at 16 pounds absolute.

Nominal cut-off = $\frac{1}{4}$ the stroke.

Assume no compression.

The actual cut-off $V_a = V_n \frac{1+C}{1+CV_n}$.

$V_n = 4$. Hence $4\frac{1+\frac{1}{9}}{1+\frac{4}{9}} = \frac{\frac{10}{9}}{\frac{13}{9}} = \frac{10}{13} \times 4 = 3$.

The mean pressure will be $115 \times \frac{1 + \text{hyp log } V_a}{V_a} =$

$115 \times 0.6993 = 80.42$ pounds.

Effective mean pressure = 80.42 − 16 = 64.42 pounds.

Let us assume that we now compress the steam to full pressure = 115 pounds.

Then $\frac{115}{16} = 7$ = rate of compression.

Then the mean pressure = 80.42 pounds as obtained before.

The effective mean pressure = $(80.42 - 16)(1+\frac{1}{9}) + \frac{115}{9}(1 - 2.95) = \frac{10}{9} \times 64.42 - \frac{224.25}{9} = 46.66$ pounds.

If there was no clearance the mean effective pressure would have been 68.59 − 16 = 52.59 pounds.

We see that the steam used in the case with full compression is the same as if there had been no clearance. The effective pressure was only 46.66 pounds. There is consequently a loss due to clearance of 52.59 pounds − 46.66 pounds, or say 5.93 pounds, or about 11 per cent.

In the first case the quantity of steam used is $\frac{13}{9}$ the volume of cylinder per stroke or one-ninth of the volume in excess of the quantity with no clearance. If with this increase of steam there was no clearance and the rate of expansion of 4 there should be an increase in the work done, and the increased work will be to the work done by the smaller quantity of steam as 13 is to 9.

We, therefore, see that the equivalent mean effective pressure is then $\frac{13}{9}$ of 52.59 or 75.96 pounds. Against 64.42 pounds, which shows a loss of 11.54 pounds or 15 per cent. This case will show the loss due to clearance, and whilst it may be considered one rarely met with in practice, yet it is sufficient to demonstrate what has been said before on this subject.

Before leaving this subject, another case will be quoted. From data of a compound engine in the author's possession we have the following:

Steam pressure, 120 pounds gauge or 135 absolute.

Receiver pressure, 25 pounds absolute.

Cut-off high-pressure cylinder, 60 per cent.

Nominal rate of expansion, 1.66.

Clearance, $\frac{1}{9}$ the cylinder volume.

We will take the first case with no compression.

Now actual rate of expansion $= 1.66 \dfrac{1+\frac{1}{9}}{1+\frac{1.66}{9}} = 1.66 \dfrac{\frac{10}{9}}{\frac{10.66}{9}} = \dfrac{1.66 \times 10}{10.66} =$ 1.55.

The mean pressure will be $135 \dfrac{1 + \text{hyp log } 1.55}{1.55} =$ $0.9292 \times 135 = 125.44$ pounds.

The effective mean pressure $= 125.44 - 25 = 100.44$ pounds.

When $\frac{3}{5} + \frac{1}{9}$ or $\frac{32}{45}$ of the volume of the cylinder of steam is used, the equivalent effective mean pressure will be $\frac{10.66}{9}$ of $97.39 = 115.35$ pounds.

The loss by clearance is, therefore, $115.35 - 100.44 = 14.91$ pounds or 13 per cent.

Now assume we compress the steam to initial pressure.

The effective mean pressure is 103.06 pounds.

The loss is, therefore, $115.35 - 103.06 = 12.29$ pounds or 10.64 per cent.

In conclusion, it is unnecessary to say the loss from clearance in a compound engine is not so serious as in a simple engine. If the clearance in the low-pressure cylinder of multiple expansion engines is large, considerable loss will occur. Otherwise, if the clearance in low-pressure cylinder is small, the losses from clearance are of no consequence. This is due to the fact, that whereas in the simple engine the cut-off is earlier, the clearance is from constructive reasons much the same. Again the ratio of clearance to volume at cut-off will be much higher. In the multiple expansion engine, the steam passing from high-pressure cylinder to the other cylinders will do more work. The exhaust steam passing to the condenser in a single cylinder condensing engine is at a higher pressure when there is clearance than when there is no clearance.

Mean Pressure in Multiple Expansion Engines

In the compound engine, if the effective mean pressure in the high pressure cylinder be divided by the ratio of the volume of low-pressure cylinder to that of the high-pressure cylinder, plus the effective mean pressure in the low-pressure cylinder the sum is termed the equivalent or referred effective mean pressure.

This referred effective mean pressure is the pressure necessary to obtain from the low-pressure cylinder alone the whole work of both cylinders.

If the effective mean pressure in the high-pressure cylinder be divided by the ratio of the volume of low-pressure cylinder to the

volume of high-pressure cylinder; the quotient is the pressure required to do the same work in the low-pressure cylinder as is effected in the high-pressure cylinder.

Thus if the ratio of $\frac{\text{L. P. Cyl.}}{\text{H. P. Cyl.}} = 4$ say.

If the effective mean pressure in high-pressure cylinder = 90 pounds.

Then the effective mean pressure in the low-pressure cylinder to do the same work as effected in high-pressure cylinder $= \frac{90}{4} = 22.5$ pounds.*

If the effective mean pressure in the high-pressure cylinder is as before 90 pounds, and the effective mean pressure in the low-pressure cylinder is 15 pounds, then the equivalent or referred effective mean pressure is equal to $\frac{90}{4} + 15 = 37.5$ pounds.

The referred effective pressure in multiple expansion engines should be the same as the effective mean pressure in a single cylinder engine having the same total rate of expansion. This, however, is never realized owing to drop in receivers, and other causes which will be taken up later.

The equivalent or referred effective mean pressure in a triple expansion engine is obtained in the same way. That is to say, the referred effective mean pressure is equal to the sum of the effective mean pressure in high-pressure cylinder divided by the ratio of the volume of low-pressure cylinder to the volume of high-pressure cylinder, plus the effective mean pressure in mean-pressure cylinder divided by the ratio of the volume of low-pressure cylinder to the volume of mean-pressure cylinder plus the effective mean pressure in low-pressure cylinder, or, placed in the form of an equation we have

If P'_m = Effective mean pressure in H. P. Cyl.
P''_m = Effective mean pressure in M. P. Cyl.
P'''_m = Effective mean pressure in 2nd M. P. Cyl.
P''''_m = Effective mean pressure in L. P. Cyl.
R = The ratio of the volume of L. P. to H. P. Cyl.
R' = The ratio of the volume of L. P. to M. P. Cyl.
R'' = The ratio of the volume of L. P. to 2nd M. P. Cyl.

Then referred effective mean pressure is $\frac{P'_m}{R} + P'''_m$ for compound.

$\frac{P'_m}{R} + \frac{P''_m}{R'} + P''''_m$ for triple expansion.

$\frac{P_m'}{R} + \frac{P''_m}{R'} + \frac{P'''_m}{R''} + P_m''''$ for quadruple expansion.

* The same reasoning applies to triple and quadruple engines.

Actual Effective Mean Pressures

The actual mean pressures in practice are less than those computed for a given initial pressure and rate of expansion.

Now the effective mean pressure is equal to the absolute initial pressure multiplied by the quotient obtained by dividing 1 plus the hyperbolic logarithm of the rate of expansion by the rate of expansion minus the absolute back pressure.

Thus if P_i =initial absolute pressure per□″ in any cylinder.
P_b =absolute back pressure per□″ in any cylinder.
R =total rate of expansion.
R_h =rate of expansion in H. P. Cyl.
R_m =rate of expansion in M. P. Cyl.
R_{m1} =rate of expansion in 2d M. P. Cyl.
R_l =rate of expansion in L. P. Cyl.

Then $P_i \times \frac{1+\text{hyp log } R}{R} - P_b$ =effective mean pressure due to the initial pressure P_i and a total rate of expansion R.*

As stated above, this pressure is, however, that which would obtain in a perfect engine, and hence is only a theoretical effective mean pressure.

In an actual engine, however, carefully designed, there will be causes of loss, and hence the actual indicator diagram will show an effective mean pressure much less than computed. The causes of loss have been treated in this chapter.

Now the ratio of the actual effective mean pressure to the theoretical effective mean pressure expresses the efficiency of the system and is termed the design or card factor.

Card Factor

The card factors vary not only for the various types of engines, but for engines of the same type, and different powers.

The following table gives a fair average:

For single engines not allowing for clearance...........	0.75	to	0.85
For single engines allowing for clearance..............	0.6	to	0.68
For compound engines not allowing for clearance	0.7	to	0.85
For compound engines allowing for clearance	0.55	to	0.7
Triple expansion engines not allowing for clearance ...	0.67	to	0.75
Triple expansion engines allowing for clearance	0.5	to	0.54
Quadruple expansion engines not allowing for clearance.	0.65	to	0.7
Quadruple expansion engines allowing for clearance....	0.55	to	0.7

In determining the card factors, it is best whenever possible to make a note of engine's performance, deducting the card factor and tabulating

* The E. M. P. for any cylinder can be found by substituting the literal quantities in the equation.

same. As an example, suppose we have a triple expansion engine, the ratio of the volume of L. P. cylinder to H. P. cylinder is 1:7.

Assume cut-off in H. P. cylinder = 70 per cent.

The total rate of expansion or R = 7 ÷ 0.75 = 0.75) 7.00 (=
075) 700.00 (9.33
675
250
225
250

Assume steam pressure 160 lbs. absolute.

Assume back pressure 5 lbs. absolute.

Now $160 \times \frac{1 + \text{hyp log } 9.33}{9.33} = 160 \times 0.3473 = 55.57$ lbs.

The mean pressure = 35.57 lbs.

The effective mean pressure = 55.57 − 5 = 50.57 lbs.

Now suppose from the indicator diagrams we have a referred effective mean pressure of 34 lbs.

The card factor would be the ratio of 34 lbs. to 50.57 lbs. = 0.672.

Now, conversely, suppose we were designing a triple expansion engine, the ratio of the volume of L. P. cylinder to H. P. cylinder = 1:7.

Cut-off in H. P. cylinder 0.75.

All conditions the same as before.

The theoretical referred effective mean pressure we found to be 50.57 lbs.

Now suppose we select a card factor of say 0.67.

Then the actual pressure would be $50.57 \times 0.67 = 33.88$, say 34 pounds.

In designing a multiple expansion engine the referred effective mean pressure is used, and after that has been determined the diam. of the low-pressure cylinder is determined.

From the remarks made before on the definition of equivalent or referred pressure, we reason about it as though the power was to be developed in the L. P. cylinder only.

With a single-cylinder engine, condensing or non-condensing, the cut-off would be total cut-off, thus with a total rate of expansion of 6 and a cut-off of 75 per cent. in the H. P. cylinder of a multiple expansion engine, the total cut-off would be $\frac{0.75}{6} = 0.125$.

The total rate of expansion, being the reciprocal of the total cut-off, would therefore be $\frac{1}{0.125} = 8$. We therefore see that with a multiple expansion engine cutting-off at 75 per cent. in the H. P. Cyl. the total

rate of expansion with a ratio of L. P. to H. P. Cyl. of 6 would be 8, while to effect this rate of expansion in a single cylinder we would cut-off at one-eighth the stroke. It is at once apparent that the great temperature range would prohibit the use of a single cylinder aside from other losses.

An example of the application of the principles enunciated in this chapter will perhaps be of benefit in aiding to comprehend fully those principles.

From data in the author's possession we will select a triple expansion engine which was designed to develop 1530 I. H. P.

The following data will be used:

Designed I. H. P. = 1530.

Steam pressure at H. P. cylinder = 150 pounds gauge.

Steam pressure at H. P. cylinder 165 pounds absolute.

Back pressure 5 pounds absolute.

Cut-off in H. P. cylinder = 0.75 = 75 per cent. of stroke.

Total rate of expansion decided upon = 8.

The theoretical referred effective mean pressure is

$$[165\# \times \frac{1+\text{hyp log } 8}{8} - 5\#].$$

But $\frac{8}{1+\text{hyp log } 8} = 0.3849.$

Theoretical mean pressure = 165 # × 0.3849 = 63.5 pounds.

Theoretical effective mean pressure = 63.5 # − 5 # = 58.5 pounds.

From diagrams of a similar engine the design factor of 0.583 was obtained.

Using this factor for our present computation we obtain:

The expected effective mean pressure = 58.5 # × 0.583 = 34.1 pounds.

As the designed horse power is to be 1530, the foot pounds of work per minute is therefore 1530 × 33000 = 50,490,000.

The stroke of piston is to be 2.75 feet = 33″.

Designed piston speed = 580.8 feet.

Revolutions = 105.6.

Computing the diameter of the L. P. cylinder we have

$$\text{Area L. P. Cyl.} = \frac{1530 \times 33000}{34.1 \times 580.8} = 2550 \text{ square inches.}$$

The nearest practical diameter is 57 inches, and the corresponding area is 2551.8 square inches.

The ratio of the volume of L. P. cylinder to H. P. cylinder must be equal to cut-off in H. P. cylinder multiplied by total rate of expansion or $0.75 = \frac{R}{8} \therefore R = 6.$

The diameter of the H. P. cylinder will be obtained, thus:

$$\text{Area H. P. cylinder} = \frac{\text{Area L. P. cylinder}}{\text{Cut-off H. P. Cyl.} \times \text{total rate of expansion}}$$

$$= \frac{2551.8\square''}{0.75 \times 8} = \frac{2551.8\square''}{6} = 425.3 \text{ square inches.}$$

The nearest practical diameter is 23.27 inches.

The area and therefore diameter of the M. P. cylinder is a subject upon which no two designers agree. It should be in the ratio of the square root of the ratio of L. P. to H. P. cylinder; this, however, gives a cylinder too large, as the temperature range is too great, and the power unequal, hence putting up excessive strains on crank shaft.

From a list of engines showing a fair distribution of power, it is found that the square root of the ratio of L. P. to H. P. cylinder is multiplied by a constant factor ranging from 1.05 to 1.1.

The diameter of the M. P. cylinder will be obtained, thus:

$$\text{Area M. P. cylinder} = \frac{\text{Area L. P. cylinder}}{F\sqrt{\text{Ratio of L. P. to H. P. Cyl.}}}$$

This engine as built had cylinders of the following dimensions:

H. P. cylinder diameter = 23½ inches.
M. P. cylinder diameter = 35 inches.
L. P. cylinder diameter = 57 inches.
Stroke common to all cylinders = 33 inches.

$$\text{The ratio of } \frac{\text{L. P.}}{\text{H. P.}} = \frac{2551.8}{433.73} = 5.88.$$

$$\text{The ratio of } \frac{\text{M. P.}}{\text{H. P.}} = \frac{962.11}{433.73} = 2.21.$$

$$\text{The ratio of } \frac{\text{L. P.}}{\text{M. P.}} = \frac{2551.8}{962.11} = 2.65.$$

The effective mean pressure H. P. Cyl. = 56.7 pounds.
The effective mean pressure M. P. Cyl. = 31.1 pounds.
The effective mean pressure L. P. Cyl. = 12.8 pounds.

The actual referred effective mean pressure is

$$\frac{56.7\#}{5.88} + \frac{31.1\#}{2.65} + 12.8\# = 34.17 \text{ pounds.}$$

The I. H. P. developed in H. P. cylinder = 432.82
The I. H. P. developed in M. P. cylinder = 526.92
The I. H. P. developed in L. P. cylinder = 574.86
The total I. H. P. = 1534.70

Note.—It is usual in designing the cylinders to be guided by temperature range, and distribution of power, etc., but as this involves a treatment which has no place in a book of this kind, as it is too abstruse, it is fully treated in the author's book on marine engine design.

Horse Power

The unit of horse power is 33,000 foot pounds per minute. This is equivalent to 33,000 pounds raised 1 foot or 1 pound raised 33,000 feet per minute.

The power to be exerted is, therefore, expressed in foot pounds. We had 1530 horse power as the desired number; we multiplied this by 33,000 foot pounds because 1 horse power is equal to 33,000 foot pounds of work per minute. Now this is the numerator of our fraction. As the horse power varies directly as the piston speed in feet per minute and as the effective mean pressure, we see that this is the denominator of our fraction.

Now the formula for horse power is $\frac{PLA2N}{33000}$.

Where P = effective mean pressure.
L = length of stroke in feet.
A = area piston in square inches.
N = number of revolutions per minute.

Now as $I.\,H.\,P. = \frac{PLA2N}{33000}$.

The area of cylinder will be given by $\frac{I.\,H.\,P \times 33000}{PL2N}$.

It must be clearly borne in mind that the effective mean pressure is the mean of the effective pressures. If the power is to be determined for each end of the cylinder separately, then the formula is $\frac{PLAN}{33000}$, and top and bottom must be added to obtain the total horse power.

Again it is readily seen that the mean pressure for each cylinder is evidently equal to the initial pressure in that cylinder, multiplied by $\frac{1+\text{hyp log of rate,}}{\text{rate}}$ where rate is the rate of expansion in that cylinder. The back pressure has to be deducted to obtain the effective mean pressure. As this is the theoretical pressure, it must be multiplied by a factor. This factor, like other factors, must be determined from the ratio of the actual effective mean pressure to the theoretical effective mean pressure. What has been said before about the reasoning on multiple expansion engines, namely, that the low pressure is treated as though all the work was to be done in that cylinder, is now sufficiently clear.

In computing the horse power developed in the cylinder or cylinders of an engine, the net area of piston is understood. That is to say, the area of piston-rod, and tail-rod, if any, must be deducted

from area of piston. As an example, suppose we have an engine of the following dimensions:

Diameter of cylinder, - - -	10 inches
Stroke of piston, - - - -	24 inches
Revolutions, - - - - - - -	100 per minute
Diameter of piston-rod, - -	2 inches
M. E. P. top from diagram, -	40 pounds
M. E. P. bottom from diagram,	36 pounds

Area of piston $= 10^2 \times .7854 = 78.54$ square inches

Therefore, I. H. P. top $= \dfrac{PLAN}{33000} = \dfrac{40 \times 2 \times 78.54 \times 100}{33000} = 19.04.$

Now I. H. P. bottom $= \dfrac{36 \times 2 \times (10^2 - 2^2) \times 0.7854 \times 100}{33000} = 16.45.$

Total I. H. P. $= 19.04 + 16.45 = 35.49.$

We can, if desirable, proceed thus:

The M. E. P. top was 40 pounds.

The M. E. P. bottom was 36 pounds.

The average M. E. P. is therefore 38 pounds.

Area of piston top $= 78.54$ square inches.

Area of piston bottom $= 78.54$ square inches $-$ 3.14 square inches $= 75.4$ square inches.

Mean area $= (78.54\square'' + 75.4\square'') \div 2 = 76.97\square''$.

The I. H. P. $= \dfrac{PLA2N}{33000} = \dfrac{38 \times 2 \times 76.97 \times 2 \times 100}{33000} = 35.49.$

If a tail rod is fitted to the piston of any cylinder, its area must be deducted from area of piston.

CHAPTER III

Combining Indicator Diagrams

Before taking up the subject of indicator diagrams in general, we will describe the method of combining same.

The object of combining the diagrams is to present in a graphical manner the losses suffered in multiple expansion engines, and to study their effects, and by proper analysis to determine the best methods for their reduction. In multiple expansion engines certain large losses appertaining to the expansive engine and not shown by the indicator diagrams are avoided. Other losses are, however, introduced which consist of those between the cylinders due to sudden expansion, wire drawing, friction, etc. It is very important to reduce all losses to the smallest possible extent; hence the value of combining and analyzing the diagrams.

The indicator diagrams which we will combine were taken from a triple expansion engine, having cylinders of the following dimensions:

Diameter of H. P. cylinder = 19″
Diameter of M. P. cylinder = 30″
Diameter of L. P. cylinder = 50″
Stroke common to all cylinders = 30″.

Fig. 20 shows the indicator diagrams from the 3 cylinders. The top and bottom diagrams are on one card.

The top diagrams only will be treated.

Taking now the diagram from high-pressure cylinder top, we divide the diagram into twenty equal spaces.* Erect ordinates perpendicular to the line of perfect vacuum. Measure the pressure at each ordinate. The pressure up to steam line and expansion line we will call plus or positive. Measure likewise the pressure between back-pressure line and vacuum line; call this pressure minus, or negative. If a scale for pressures corresponding with spring used in instrument when diagrams were taken is not at hand, we can measure each ordinate in inches and convert same into pounds, per square inch. Thus, if the ordinate is 1¾″ long and the scale used was, say, 80 pounds, the pressure would be $1.75 \times 80 = 140$ pounds per square inch.

In using 20 ordinates the work is more tedious, but the result amply repays for any extra work, as the enlarged diagram is more accurate. After having divided the high-pressure diagram as described, we pro-

* Some prefer to divide diagram into 10 equal spaces; 20, however, are more accurate.

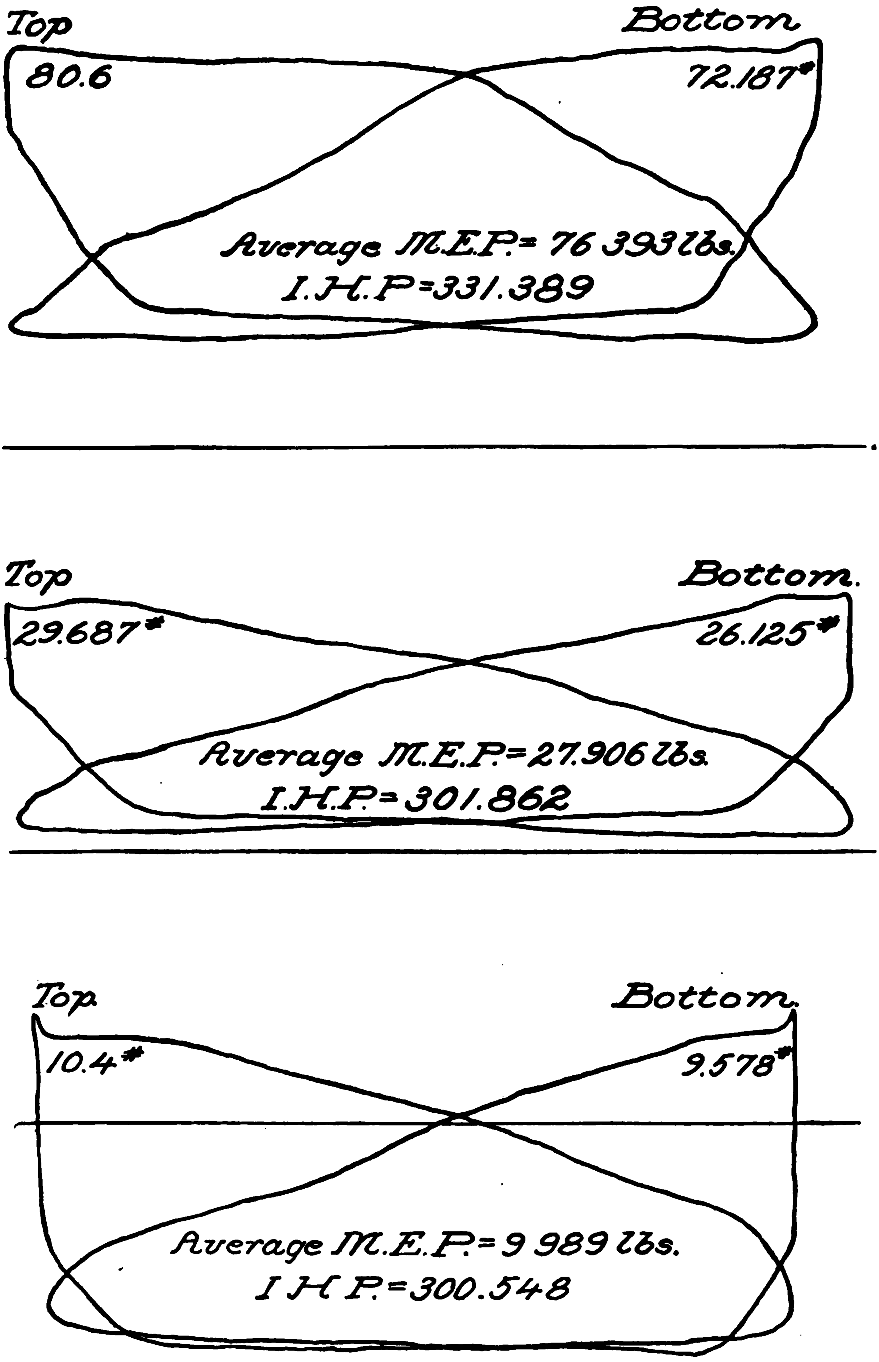

Top
Bottom
80.6
72.187#
Average M.E.P.= 76 393 lbs.
I.H.P=331.389
Top
Bottom.
29.687#
26.125#
Average M.E.P.=27.906 lbs.
I.H.P.=301.862
Top
Bottom.
10.4#
9.578#
Average M.E.P.=9 989 lbs.
I H P.=300.548

Fig. 20

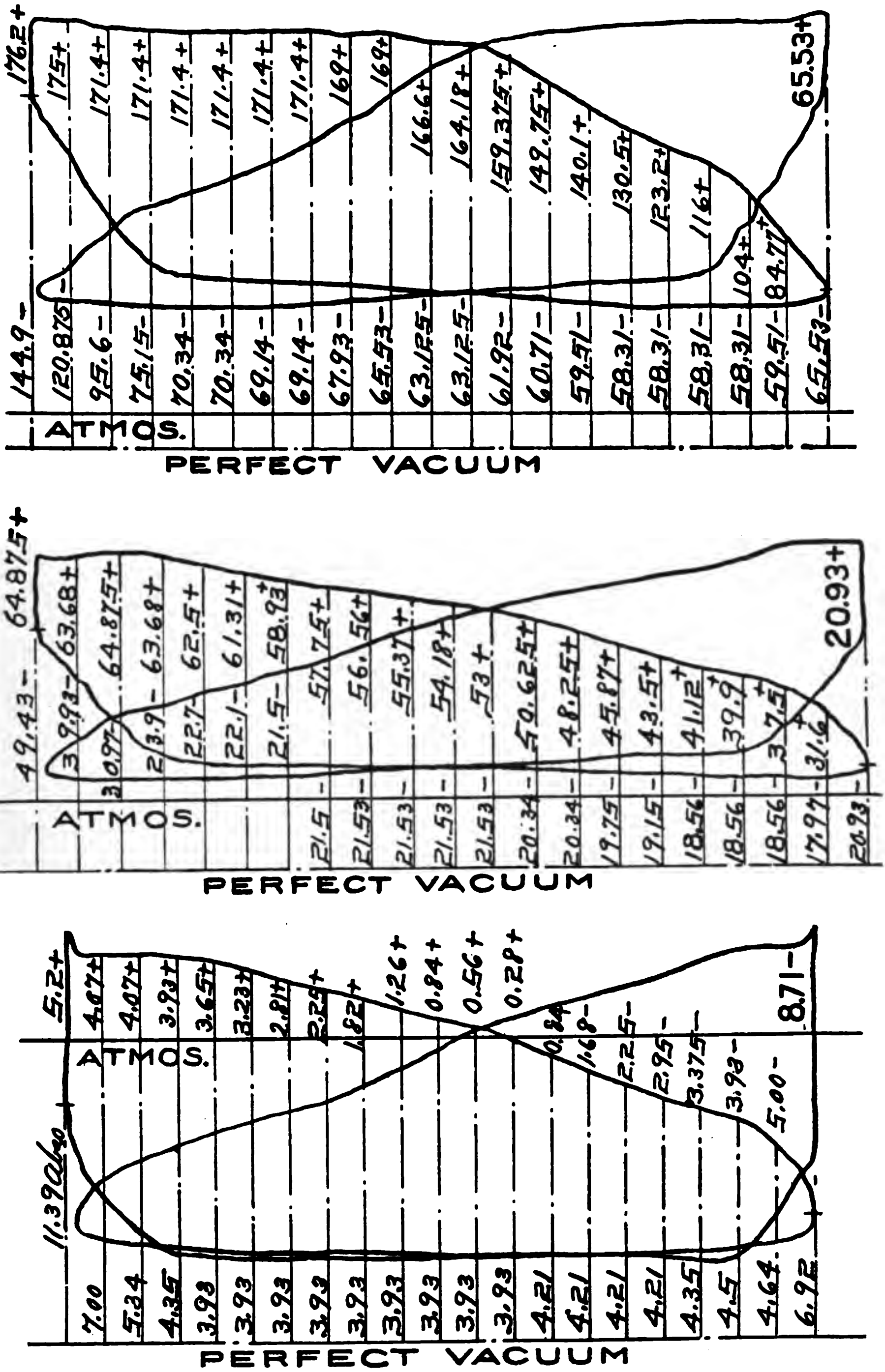

ATMOS.
PERFECT VACUUM
ATMOS.
PERFECT VACUUM
ATMOS.
PERFECT VACUUM

Fig. 21

ceed to treat the diagram from M. P. cylinder and L. P. cylinder, in precisely the same manner. Fig. 21 shows the diagrams of Fig. 20 divided, and the corresponding pressure inserted.

The combined diagram is shown on plate 2.

The method of construction is as follows: Draw a horizontal line OX, and a vertical line OY, intersecting OX in O. The horizontal line OX is a line of volume; the vertical line OY is a line of pressure, or perhaps more correctly the line on which pressures are set off.

The line OX is also the line of perfect vacuum. In combining diagrams the volumes of the different cylinders are set off in their proper volumetric ratios; whilst the pressures are all set off to the same scale.

For pressures we will use a scale of 10 pounds to the inch; thus every inch in height on line OY represents 10 pounds pressure per square inch on piston.

Set off from O on OY pressures up to the absolute boiler pressure, thus 0, 10, 20,......187 as shown.

The boiler pressure at the time these diagrams were taken was 172 pounds gauge or 187 pounds absolute. Line OY is not only a line of pressure, but it is also the line from which the clearance in each cylinder is measured. We must know the volumetric clearance in each cylinder before we can combine the diagrams. As mentioned in chapter II, this is a very difficult undertaking after engines are erected and in the ship. It is then necessary to obtain this information from the builders. The clearances for this engine was determined from the drawings of the cylinders and was found to be as follows:

Volumetric clearance H. P. cylinder = 14 per cent.
Volumetric clearance M. P. cylinder = 14 per cent.
Volumetric clearance L. P. cylinder = 9 per cent.

From the line OY set off parallel with OX, and to the right a distance equal to the clearance in H. P. cylinder. Before doing this, however, we must decide upon what length to make the H. P. cylinder diagram. The length of diagram is entirely optional and depends upon the whim of the engineer. 2 inches makes a good length of diagram, as then each ordinate is $\frac{1}{10}''$ apart, that is to say, the interval is 0.1 inch.

We will adopt a length of 2 inches. Now 14 per cent. of 2 inches is equal to $0.14 \times 2 = 0.28$ inch. Set off, therefore, from OY a distance of 0.28 inch and draw a vertical line parallel with OY. Draw a horizontal line parallel with OX, at a height corresponding to the initial absolute pressure in H. P. cylinder, which is in this case 176.2 pounds absolute. Set off a distance from OY on the horizontal line mentioned, a distance of 2.28 inches, or 2 inches from the clearance line. Now divide the

2 inches into 20 equal parts, drawing ordinates parallel with OY. Set off on these ordinates the pressures corresponding to the similar ordinates on the indicator diagram (Fig. 21).

After these pressures are all set off on their respective ordinates for both the forward and return stroke, we trace a curve through the points and obtain the contour of diagram. It is best in all cases when dealing with pressures to deal with absolute pressures, because pressures are set off from vacuum. In taking pressures from the diagram it it better to take from vacuum line also. This line can be drawn on each card, by setting off below the atmospheric line a distance corresponding to 15 pounds to the scale with which diagram was taken.

Intermediate Cylinder Diagram

The diameter of the M. P. cylinder is 30 inches.

The area of a 30-inch cylinder is 706.86 square inches.

The diameter of the H. P. cylinder is 19 inches.

The area of a 19-inch cylinder is 283.53 square inches. The ratio of the volume of M. P. to H. P. is, therefore, $706.86 \div 283.53 = 2.49$.

The high pressure diagram having been made 2 inches in length, the length of the M. P. diagram will, therefore, be $2.49 \times 2 = 4.98$ inches.

The clearance in M. P. cylinder was found to be 14 per cent.; therefore, 14 per cent. of $4.98 = 4.98'' \times 0.14 = 0.697$ inch. Set off from OY a distance equal to 0.697 inch, draw a line parallel with OY. Draw a horizontal line parallel with OX, at a height corresponding to the initial absolute pressure, in this cylinder, which in this case is 64.875 pounds absolute. Set off from OY on the horizontal line just described, a distance of 5.677 inches or 4.98 inches from the clearance line. Now divide the 4.98 inches into 20 equal parts, drawing ordinates parallel with OY. Set off on these ordinates the pressures corresponding to the similar ordinates on the indicator diagram (Fig. 21). After these pressures are all set off on their respective ordinates, as explained for the H. P. diagram. and the curves drawn in, we have the contour of the M. P. cylinder diagram.

Low-pressure Diagram

The diameter of the L. P. cylinder is 50 inches.

The area of a 50-inch cylinder is 1963.5 square inches.

The diameter of the H. P. cylinder is 19 inches.

The area of a 19″ cylinder is 283.53 square inches.

The ratio of the volume of L. P. to H. P. is therefore, $1963.5 \div 283.53 = 6.92$.

The high-pressure diagram having been made 2 inches in length, the length of the L. P. diagram will therefore be $6.92 \times 2 = 13.84$ inches.

The clearance in L. P. cylinder was found to be 9 per cent.; therefore, 9 per cent. of $13.84 = 13.84 \times 0.09 = 1.24$ inches. Set off from OY a distance equal to 1.24 inches; draw a line parallel with OY. Draw a horizontal line parallel with OX, at a height corresponding to the initial absolute pressure in this cylinder, which is in this case 20.2 pounds absolute. Set off from OY on the horizontal line just described, a distance of 15.08 inches, or 13.84 inches from the clearance line. Now divide the 13.84 inches into 20 equal parts, drawing ordinates parallel with OY. Set off on these ordinates the pressures corresponding to the similar ordinates on the indicator diagram, Fig. 21. After these pressures are all set off on their respective ordinates as explained for the H. P. and M. P. diagrams, and the curves drawn in, we have the contour of the L. P. cylinder diagram.*

We now have the three diagrams drawn to the same scale of pressures, and each diagram set out in its proper volumetric ratio, and with their proper clearances.

The next step is to draw the $PV = C$ curve.

The method of doing this has been described in a previous chapter, and need not be treated here. Any of the curves can be drawn, and they are of interest, and should be practiced by the student.

Drawing the curve $PV = C$ through the point of cut-off as shown, we note that, producing this curve to the maximum initial pressure, the cut-off is slightly reduced. This is known as the reduced cut-off, for we see that the cut-off on the indicator diagram of H. P. cylinder is 59 per cent. This is the nominal cut-off. The actual cut-off is nominal cut-off + clearance $= 0.59 + 0.14 = 73$ per cent. The reduced cut-off should be $\frac{(0.59+0.14)\times 161.52}{176.2} = 0.67$ or 67 per cent.

Measuring the combined diagram we see that it measures just 67 per cent. for $1\frac{1}{16}'' \div 2 = 53$ per cent.

$0.53 + 0.14 = 0.67$ or 67 per cent.

161.52 pounds is the cut-off pressure.

176.2 pounds is the initial pressure on H. P. piston.

Back Pressure Line

The assumed back pressure is 4 pounds absolute. From O on OY, set off a distance equal to 4 pounds, draw a horizontal line parallel with the perfect vacuum line OX.

Atmospheric Line

The atmospheric line should be drawn after pressure and vacuum lines are established. Therefore, from O on OY, set off a distance

* It may be found by some to be more desirable to work from the atmospheric line for H. P. and M. P. diagrams and above and below atmospheric line for L. P. diagram. This is optional.

equal to 15 pounds, draw a horizontal line parallel with the perfect vacuum line OX.

Looking at the combined diagrams, plate 2, we note that there is a drop of 10.8 pounds between boiler and piston of H. P. cylinder.

The boiler pressure was 187 pounds absolute.

The initial pressure by indicator diagram is 176.2 pounds absolute. Therefore, $187-176.2=10.8$ pounds.

There is also a drop between the initial pressure and cut-off pressure. The cut-off pressure is 161.52 pounds, and the difference between 176.2 pounds and 161.52 pounds $=14.68$ pounds.

The pressure in first receiver was 67 pounds. The initial pressure in M. P. cylinder was 64.875 pounds.

There is a drop in this receiver of 67 pounds $-$ 64.875 pounds $=2.125$ pounds.

The pressure in second receiver was 21 pounds.

The initial pressure in L. P. cylinder was 20.2 pounds.

There is a drop in this receiver of 21 pounds $-$ 20.2 pounds $=0.8$ pounds.

The theoretical diagram is that represented by OY, OX, and the curve $PV=C$.

The effective mean pressure of the ideal diagram is obtained as follows:

The initial steam pressure is 176.2 pounds absolute.

The reduced cut-off was 67 per cent. This is an actual and not a nominal cut-off.

The ratio of the volume of the L. P. cylinder to the H. P. cylinder is 6.92.

Now $0.67=\frac{6.92}{X}$. Therefore, the total rate of expansion

$X=6.92\div 0.67=10.32$.

Now $\frac{1+\text{hyp log } 10.32}{10.32}=0.3224$.

The theoretical mean pressure $=176.2\times 0.3224=56.8$ pounds.

The theoretical effective mean pressure $=56.8$ pounds $-$ 4 pounds $=$ 52.8 pounds.

The effective mean pressure shown by H. P. diagram $=80.6$ pounds.

The effective mean pressure shown by M. P. diagram $=29.687$ pounds.

The effective mean pressure shown by L. P. diagram $=10.4$ pounds.

Then the effective mean pressure referred is as before equal to $\frac{80.6}{6.92}+\frac{29.687}{2.77}+10.4=11.64$ pounds $+$ 10.71 pounds $+$ 10.4 pounds $=32.75$ pounds.

Now, as explained before, the card factor is a ratio, and represents the percentage of returns for investment. The card factor in this case is, therefore, $32.75 \div 52.8 = 0.62$. That is to say, the actual pressure is 62 per cent. of the theoretical. If the theoretical diagram is to be considered from initial pressure H. P. cylinder to perfect vacuum, then the card factor would be $32.75\# \div 56.8\# = 0.576$.

In all engineering investigations, accuracy should be the prime factor. Not only in the analysis and computations, but the instruments with which the data is obtained should be accurate, and should the instrument be in error, this error must be determined and allowed for. It will be found profitable, after all measurements of the diagrams have been made and recorded, to determine the effective mean pressures, from the measurements made, before combining, as the measurements are many, and having previously found the effective mean pressure of the diagrams by planimeter, it is a good check.

An example will make these remarks clear.

The effective mean pressure of the top indicator diagram of H. P. cylinder was found to be 80.6 pounds; from the ordinates we have 80.18 pounds. It is shown by Fig. 21 that measuring between the limits of the diagram the following pressures are obtained.

1st	Ordinate	31.3	pounds.
2d	"	54.125	"
3rd	"	75.8	"
4th	"	96.25	"
5th	"	101.06	"
6th	"	101.06	"
7th	"	101.06	"
8th	"	101.06	"
9th	"	101.07	"
10th	"	103.47	"
11th	"	103.475	"
12th	"	101.055	"
13th	"	97.455	"
14th	"	89.04	"
15th	"	80.59	"
16th	"	72.19	"
17th	"	64.97	"
18th	"	57.69	"
19th	"	45.69	"
20th	"	25.26	"
21st	"	0.	"
		Sum = 1603.67	

And $1603.67 \div 20 = 80.18$ pounds effective mean pressure. Showing a difference $= 80.6 - 80.18 = 0.42$ pounds, or .5 per cent. That is $\frac{1}{2}$ of 1 per cent. less.

Treating the M. P. and L. P. diagrams in a similar manner we obtain for the top diagram of M. P. cylinder 29.25 pounds. The effective mean pressure of the same diagram by planimeter is 29.687 pounds. Showing a difference of $29.687 - 29.25 = 0.437$ pounds, or 1.4 per cent. less.

For the top diagram of L. P. cylinder 10.81 pounds. The effective mean pressure of the same diagram by planimeter is 10.4. Showing a difference of $10.81 - 10.4 = 0.41$ pounds or nearly 4 per cent. greater.

This is sufficient to prove the accuracy of the different pressures. It will be noticed that in each diagram of the combined diagram, the effective mean pressure is inserted. Each diagram was carefully traced over with the planimeter and the pressures inserted obtained.

It may have been noted that the remarks upon the combined diagram took no account of the clearance in the L. P. cylinder. The diagrams and the combined diagrams, fig. 21a, are from the same engine as those shown on plate 2, but at a different time. Now taking into consideration the clearance in L. P. cylinder, our computations would be as follows: The nominal cut-off in H. P. cylinder is 75 per cent. The clearance in H. P. cylinder is equal to 14 per cent. of the cylinder volume.

The initial pressure as shown by H. P. cylinder diagram is 165.38 pounds absolute.

The pressure at cut-off H. P. cylinder as shown by diagram is 157.88 pounds absolute.

The equivalent cut-off from measurement is 84.5 per cent.

Thus nominal equivalent cut-off from measurement $= 70.5$ per cent. $70.5 + 14 = 84.5$ per cent.

The actual equivalent cut-off by computation is

$$\frac{(75+14) \times 157.88}{165.38} = 0.849 = 84.9 \text{ per cent.}$$

Initial volume for expansion is therefore 84.9 per cent.

The finial volume will therefore be $(100+9) \times 6.92$ where $6.92 =$ the ratio of $\frac{\text{L. P.}}{\text{H. P.}}$

Clearance in L. P. cylinder $= 9$ per cent. of the cylinder volume. Now $109 \times 6.92 = 754.28$

$$\text{The cut-off is therefore} = \frac{\text{initial volume}}{\text{finial volume}} = \frac{84.9}{754.28} = 0.112.$$

$$\text{The total rate of expansion} = \frac{1}{R} = \frac{1}{0.112} = 8.92.$$

If we take and divide the distance OX into volumes equal to OU, we see that it contains OU just 8.92 times. By the shorter method, as previously described, we have

Equivalent cut-off $=0.845$.

Ratio $\frac{\text{L. P.}}{\text{H. P.}}=6.92$.

Total ratio of expansion $=\frac{6.92}{0.845}=8.18$.

The mean pressure per pound for $8.920=.358$.

The mean pressure per pound for $8.180=.3759$.

Taking initial pressure 165.38 pounds in both cases, we have

$165.38\times0.358 \;\; =59.2$ pounds.

$165.38\times0.3759=62.16$ pounds.

Deducting 4 pounds back pressure in both cases, we have for effective mean pressure:

$59.2-4=55.2$ pounds.

$62.16-4=58.16$ pounds.

The difference $=2.96$ pounds, or 5 per cent.

The effective referred mean pressure from diagrams $=33.7$ pounds.

The card factor in the former case is $\frac{35.91}{55.2}=0.65$.

The card factor in the latter case is $\frac{35.91}{58.16}=0.617$.

Some designers do not deduct an assumed back pressure, treating the area between initial pressure and a vacuum.

The card factor then becomes in the first case: $\frac{35.91}{59.2}=0.6$.

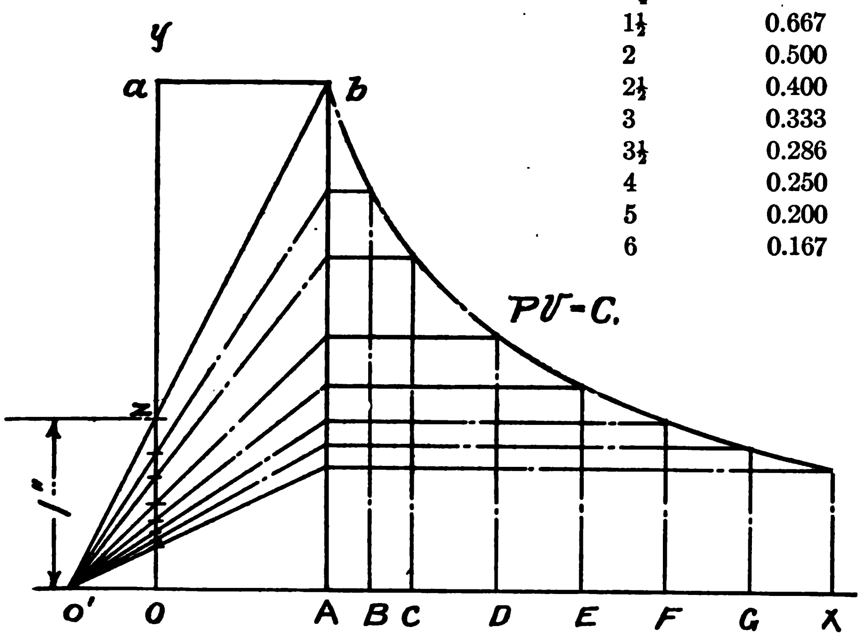
In the latter case the card factor is: $\frac{35.91}{62.16}=0.577$.

It is thus seen that when the first value is taken or the first method, the cylinders would be slightly smaller than with the second method. That is to say, in designing with a referred, effective mean pressure the cylinders would be slightly smaller with the clearance in L. P. cylinder taken into consideration. It is, therefore, better to deal with the actual values from similar engines, and in computing the effective theoretical mean pressure from the combined diagram the clearance in L. P. cylinder must b onsidered. Computing from actual data the card factor for several types of engines, the following gives a fair mean when determining the mean referred pressure without taking a theoretical back-pressure into consideration.

COMPOUND ENGINES

Large engines up to 100 revolutions per minute	0.6 to 0.68
Small engines	0.5 to 0.6
Triple expansion 3-cylinder engines	
Mercantile ships	0.55 to 0.58
Triple expansion 4-cylinder engines	0.5 to 0.54
Quadruple expansion	0.52

Volume	Decimal
1¼	0.800
1½	0.667
2	0.500
2½	0.400
3	0.333
3½	0.286
4	0.250
5	0.200
6	0.167



Equilateral Hyperbola

Fig. 22

It is absolutely necessary to exercise the greatest care in not only taking diagrams, but in computing the data, for unless the data is reliable it is simply a waste of time to analyze results. The value to the designer as well as to the practical engineer of the information to be derived from the indicator diagram cannot be over-estimated.

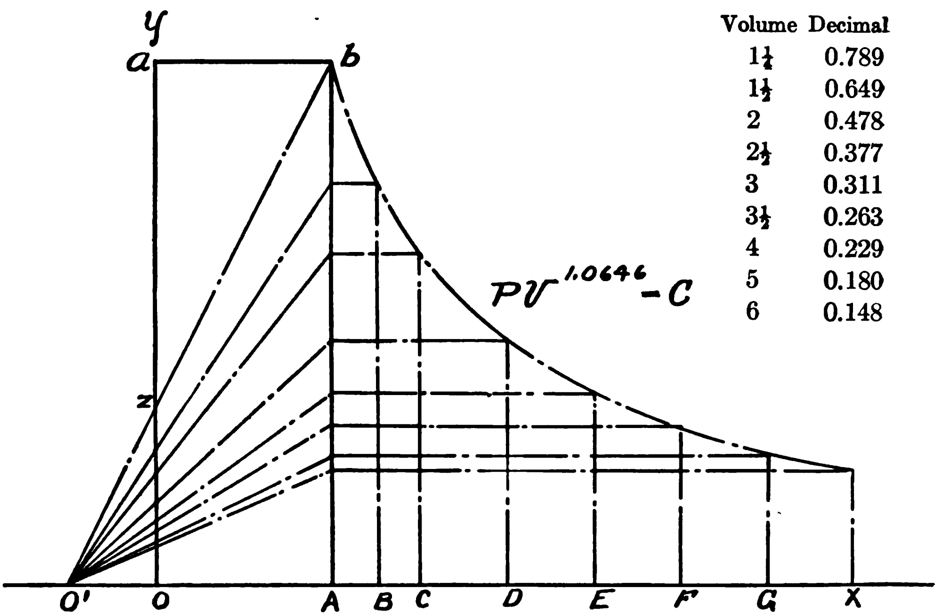
Before closing this subject we will consider some other curves, and describe the method of constructing them. It was at one time customary to plot what was termed the saturation curve when cards were combined. Others treated the PV=C curve as the curve of saturation.

The PV=C curve is and has been repeatedly referred to as the theoretical curve of expansion. In previous remarks, we see how absurd such reference is. The equation to the saturation curve is $PV^k=C$.

Now the exponent k for this curve is 1.0646, whilst the exponent for the hyperbola is 1.

The adiabatic curve is $PV^k=C$. The exponent k for this curve is 1.13.

It is interesting to plot these curves on a combined indicator diagram, to see their variations and peculiar features, and the exercise is highly instructive.



Volume	Decimal
1¼	0.789
1½	0.649
2	0.478
2½	0.377
3	0.311
3½	0.263
4	0.229
5	0.180
6	0.148

Saturation Curve.

Fig. 23

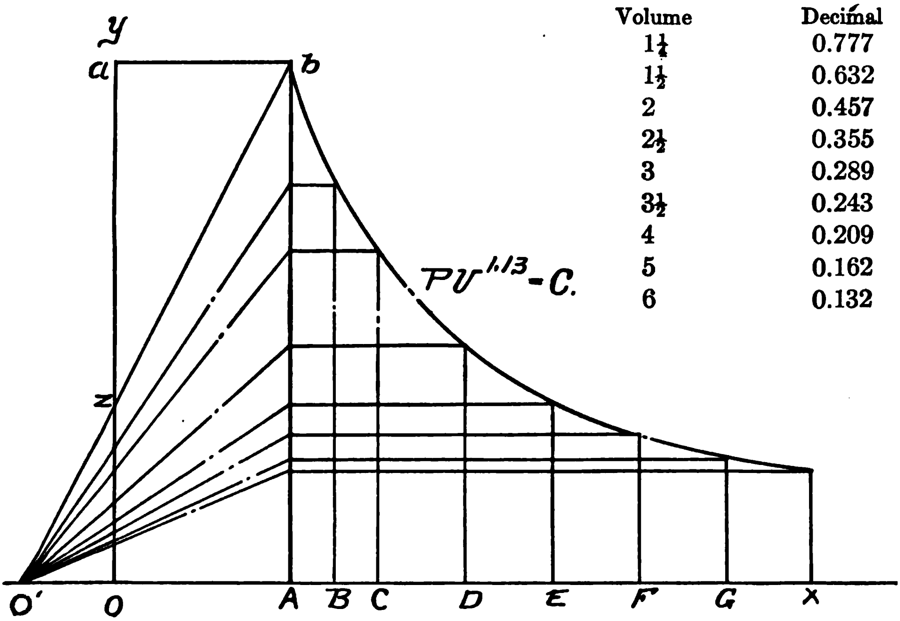
Curves of Expansion

With each figure there is given a table of the constants used in constructing the respective curves.

Fig. 22 shows a practical method of plotting the PV=C curve, and its construction is as follows: Let OY represent the absolute initial pressure; from O set off on OY a distance of 1 inch represented by OZ. Now set off on the line OX a distance equal to the volume up to cut-off. Complete the rectangle OYBA.

Draw a diagonal line from B passing through Z, and produce same to pass through O′ on the line of perfect vacuum produced. Set off on OX, a distance OB=1¼OA, OC=1½OA, OD=2OA, OE,=2½OA, OF=3OA, OG=3½OA, etc.

Now from O set off on OY a distance=0.8 inch for 1¼ vols. 0.667 for 1½ vols., 0.5 for 2 vols., 0.4 for 2½ vols., 0.333 for 3 vols., 0.286 for 3½ vols. and 0.25 for 4 vols. Now pass diagonals through the corresponding points from O′ intersecting AB. From these points of intersection pass horizontal lines parallel with OX. The horizontals intersecting the ordinates erected on OX, as shown, locate points of the



Volume	Decimal
1¼	0.777
1½	0.632
2	0.457
2½	0.355
3	0.289
3½	0.243
4	0.209
5	0.162
6	0.132

Adiabatic Curve

Fig. 24

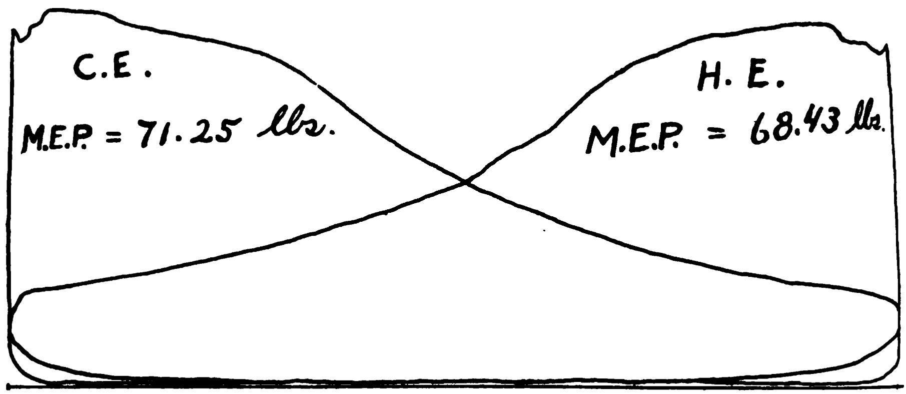
curve; passing a fair curve through these points gives us a curve known as the equilateral hyperbola, or PV=C curve. Taking the combined indicator diagrams, the volume is 73 per cent. and proceeding as just described we obtain the curve as there plotted.

Fig. 23 shows the saturation curve. This curve is constructed in precisely the same manner as the PV=C curve. The decimal corresponding to the volume is given in figure.

Fig. 24 shows the adiabatic curve of expansion. Constructed the same as explained for the two preceding curves. The decimal corresponding to the volume is given in the figure.

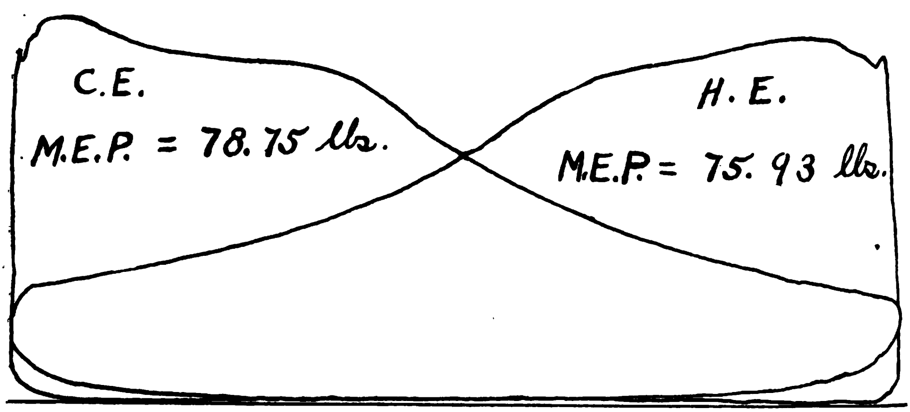
Indicator Diagrams

The following four diagrams are taken from the steam tug "Baltic." Diameter cylinder 16 inches; stroke of piston 16 inches.



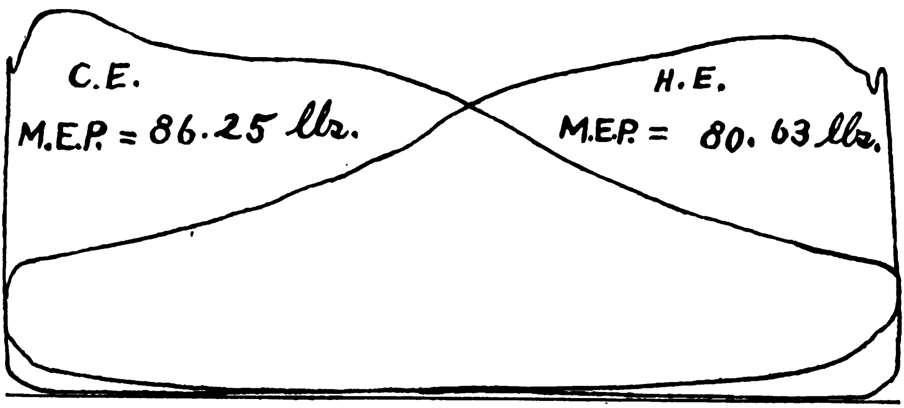
Average M. E. P., 69.84 lbs.

6″ Cut-off. Steam, 120 lbs. 118 Revo.
133.561 I. H. P.
Scale of Spring, 60



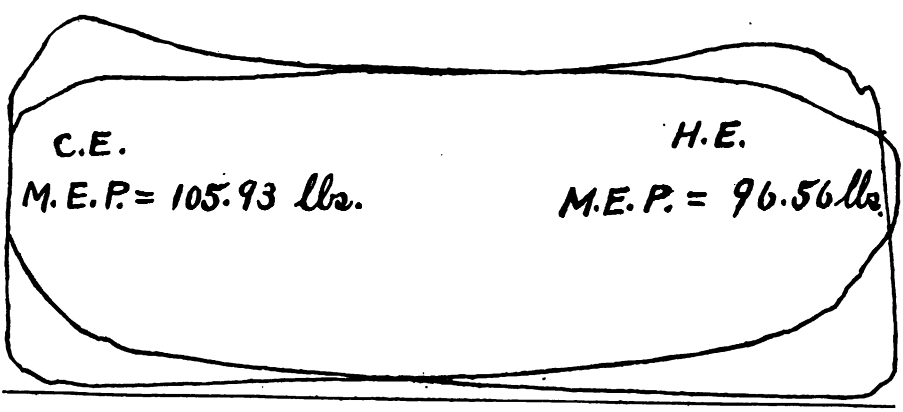
Average M. E. P., 77.34 lbs.

7″ Cut-off Steam, 120 lbs. 120 Revo.
150.325 I. H. P.
Scale of Spring, 60



Average M. E. P., 83.44 lbs.

8″ Cut-off Steam, 120 lbs. 126 Revo.
170.317 I. H. P.
Scale of Spring, 60



Average M. E. P., 101.245 lbs.

Full Stroke of Valve Steam, 120 lbs. 132 Revo.
216.502 I. H. P.
Scale of Spring, 60

The following diagrams are from the first compound engine built in America.

This engine has cylinders of the following dimensions

High pressure cylinder 24 inches.

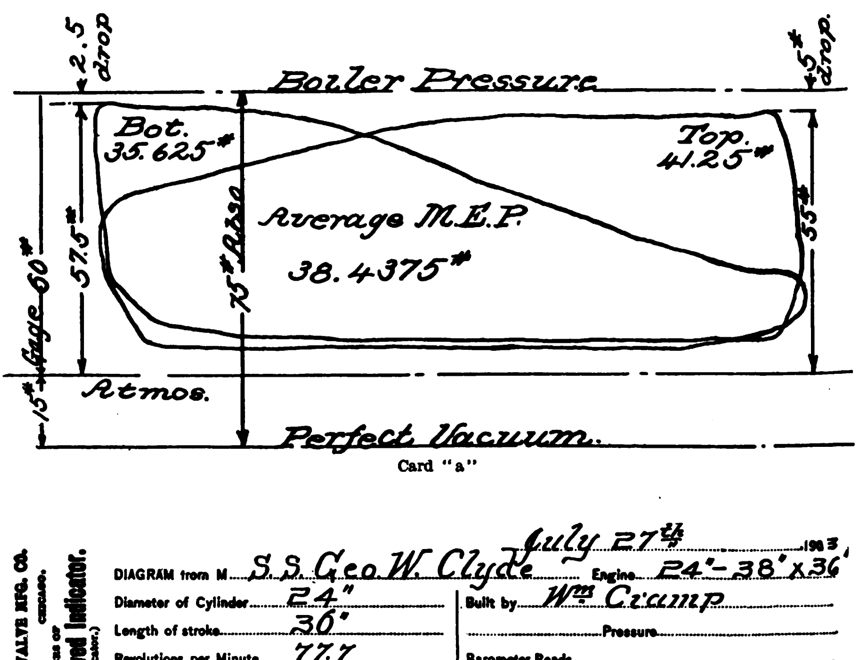
Low pressure cylinder 38 inches.

Stroke common to both 36 inches.

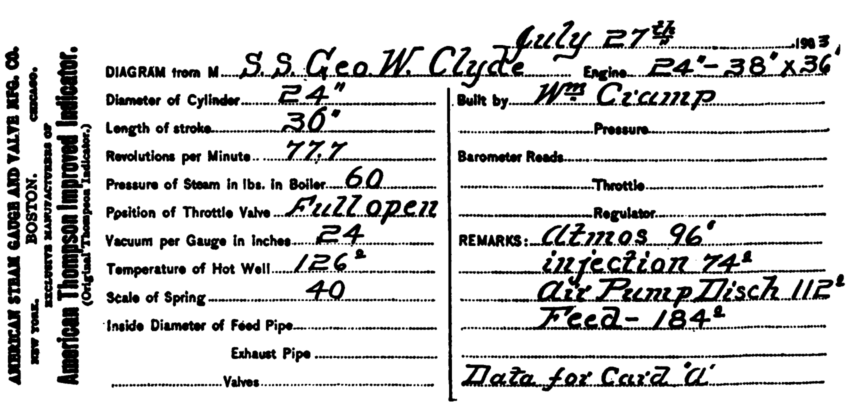
In the first chapter I stated that the defects incident to long indicator pipes would be discussed later.

Diagrams "a" and "b" are from the H. P. and L. P. cylinders respectively. The indicator pipe was arranged as shown in inset facing page 66 in fig. 1.

Looking at card "a,"



Card "a"



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July 27th 1903

DIAGRAM from M. S. S. Geo W. Clyde Engine 24"-38"x36"

Diameter of Cylinder 24"	Built by Wm Cramp
Length of stroke 36"	Pressure
Revolutions per Minute 77.7	Barometer Reads
Pressure of Steam in lbs. in Boiler 60	Throttle
Position of Throttle Valve Full open	Regulator
Vacuum per Gauge in Inches 24	REMARKS: Atmos 96°
Temperature of Hot Well 126°	injection 74°
Scale of Spring 40	Air Pump Disch 112°
Inside Diameter of Feed Pipe	Feed - 184°
Exhaust Pipe	
Valves	Data for Card "a"

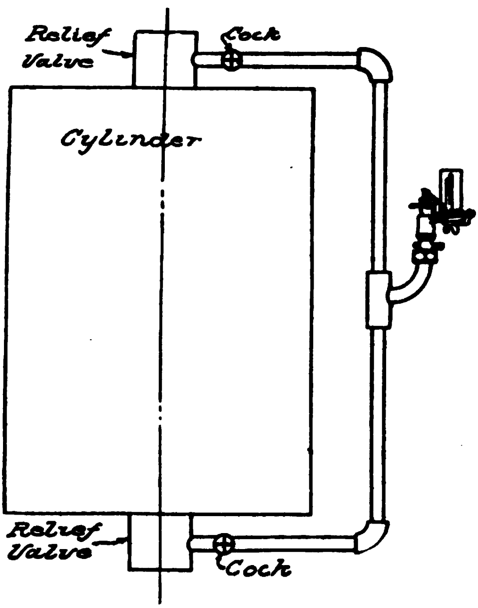

Relief
Valve
Cock
Cylinder
Relief
Valve
Cock

Fig .1

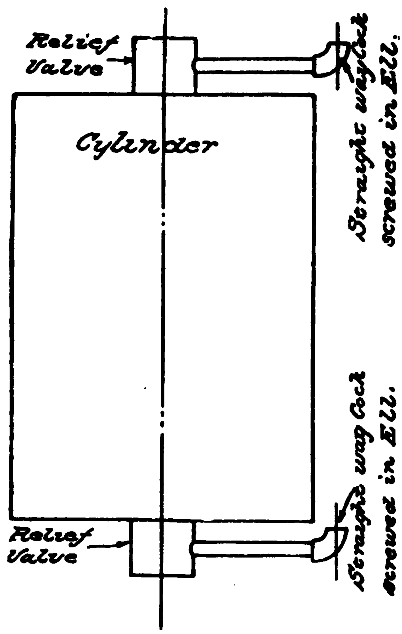
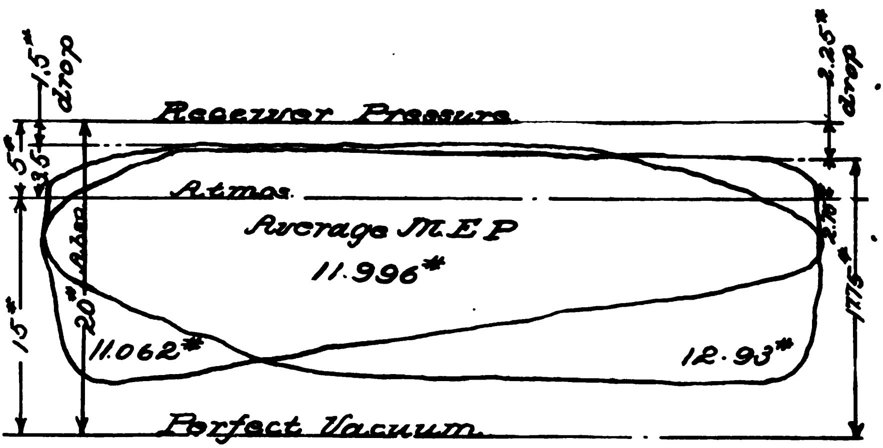

Relief
Valve
Cylinder
Straight way cock
screwed in Ell.
Relief
Valve
Straight way Cock
screwed in Ell.

Fig. 2

we see the drop between boiler and H. P. piston is 5 lbs. for top, 2.5 lbs. for bottom. The initial steam pressure top is 55 lbs. gauge or 70 lbs. absolute. For the bottom the initial pressure is 57.5 lbs. gauge or 72.5 lbs. absolute. The absolute steam pressure is 75 lbs. The M. E. P. top is 41.25 lbs. M. E. P. bottom is 35.625 lbs. Giving a difference between top and bottom of 5.625 lbs. The average M. E. P. is 41.25 # +35.625 # = 76.875 lbs. 76.875÷2 = 38.4375 lbs.

For card "b,"



Card "b"

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July 27th 1903

DIAGRAM from S. S. Geo W. Clyde Engine 24"-38"x36"

Diameter of Cylinder 38"	Built by Wm Cramp
Length of stroke 36"	Pressure
Revolutions per Minute 77.7	Barometer Reads
Pressure of Steam in lbs. in Boiler 60	Throttle
Position of Throttle Valve Full Open	Regulator
Vacuum per Gauge in inches 24	REMARKS: Atmos 96°
Temperature of Hot Well 126°	injection 74°
Scale of Spring 12	Air Pump Disch 112°
Inside Diameter of Feed Pipe	Feed 184°
" " Exhaust Pipe	
Valves	Data for Card "b"

we have a receiver pressure of 5 lbs. gauge or 20 lbs. absolute. The drop in receiver is for top 2.25 lbs. and 1.5 lbs. for bottom. The initial steam pressure top is 2.75 lbs. gauge of 17.75 lbs. absolute. For the bottom the initial pressure is 3.5 lbs. gauge or 18.5 lbs. absolute.

The M. E. P. top is 12.93 lbs. M. E. P. bottom is 11.0625 lbs., giving a differenfie of 1.8675 lbs. The average M. E. P. is 12.93 # + 11.062 # = 23.992 lbs. 23.992 ÷ 2 = 11.996 lbs.

The M. E. P. top is 12.93 lbs. M. E. P. bottom is 11.0625 lbs., giving a difference of 1.8675 lbs. The average M. E. P. is 12.93# + 11.062# = 23.992 lbs. 23.992 ÷ 2 = 11.996 lbs.

The constant for the H. P. cylinder $= \dfrac{\text{PLA2N}}{33000}$.

Let the M. E. P. pressure = 1 pound.

Piston speed in feet = 1 foot per minute.

Then the constant for 1 lb. M. E. P and one foot of piston speed =

$$\frac{1\times1\times452.39\times2\times N}{33000}=\frac{904.78}{33000}=0.02741.$$

The constant for L. P. cylinder $=\dfrac{1134.1\times2}{33000}=\dfrac{22682}{33000}=0.06873.$

The average M. E. P. H. P. cylinder = 38.4375

The average revolutions = 77.7

The stroke of piston = 3 feet.

The indicated horse power developed in H. P. cylinder is, therefore, $C\times$M. E. P.$\times N\times L=0.02741\times38.4375\times77.7\times3=245.571$ horse power.

The indicated horse power developed by L. P. cylinder is, therefore, $C\times$M. E. P.$\times N\times L=0.06873\times11.996\times77.7\times3=192.189$ horse power.

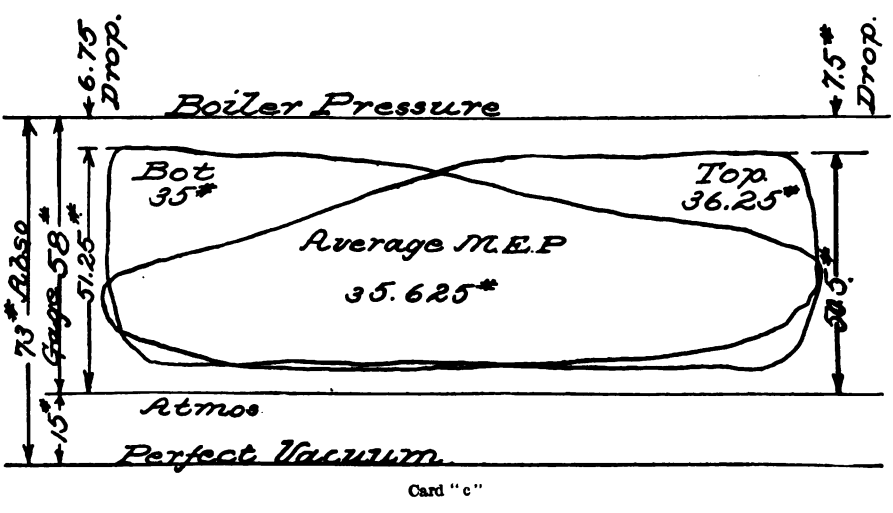
The collective I. H. P. = 245.571 + 192.189 = 437.76.

The ratio of cylinder capacities = area L. P. cylinder ÷ area H. P. cylinder = 1134.1 ÷ 452.59 = 2.56.

The aggregate equivalent M. P. referred to L. P. piston is, therefore,

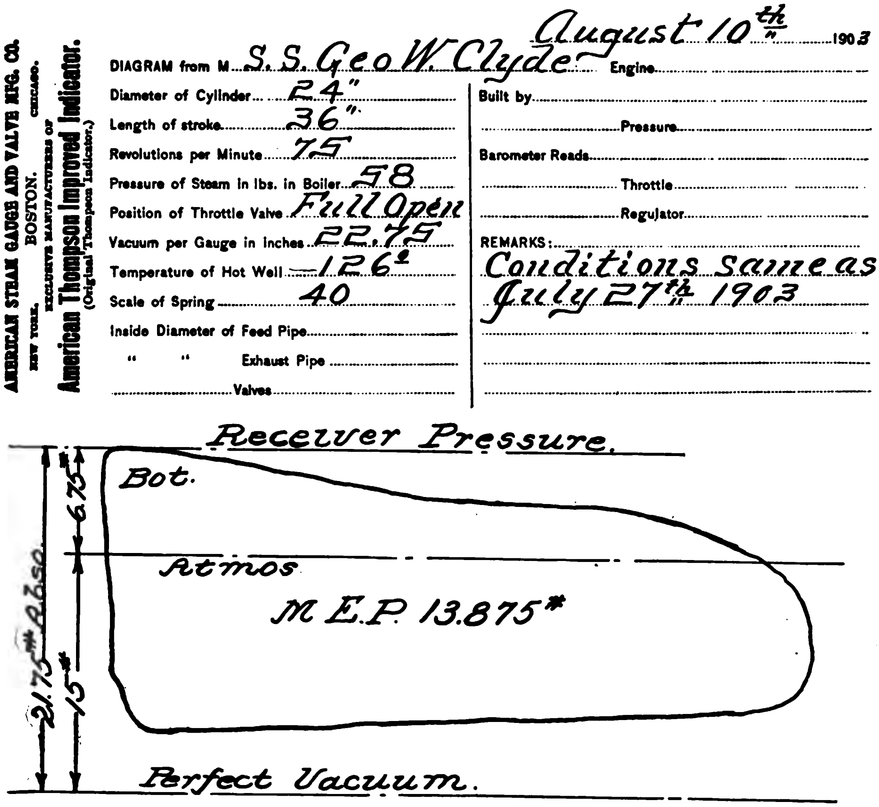
$$\frac{\text{M. E. P. H. P. Cyl.}}{\text{Ratio }\frac{\text{L. P.}}{\text{H. P.}}}+\text{M. E. P. L. P. Cyl.}=\frac{38.4375}{2.56}+11.996=$$

15.01# + 11.996# = 27 lbs.

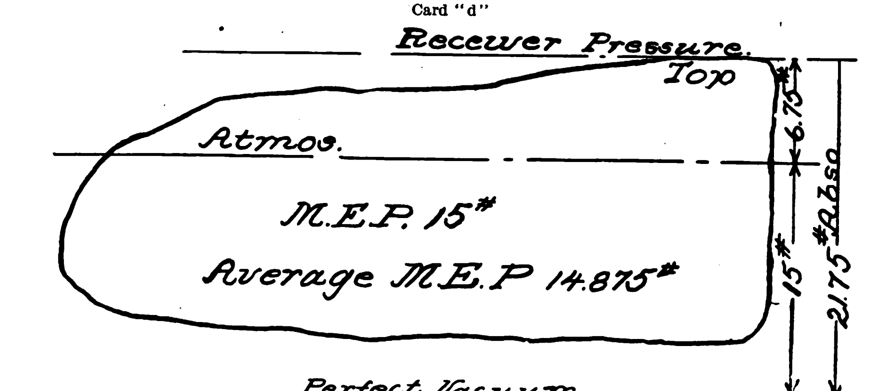


Card "c"

Diagrams "c," "d" and "e" are from the same engine but with short connections (see fig. 2 of inset facing page 66).



Card "d"



Card "e"

Taking diagram "c" we see that the drop between boiler and H. P. piston is 7.5 lbs. for top, 6.75 for bottom. The initial steam pressure

top is 50.5 lbs. or 65.5 lbs. absolute. For the bottom the initial pressure is 51.25 lbs. gauge or 66.25 lbs. absolute. The absolute steam pressure is 73 lbs. The M. E. P. top is 36.25 lbs., M. E. P. bottom is 35 lbs., giving a difference between top and bottom of 1.25 lbs.

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August 10th 1903

DIAGRAM from M S. S. Geo W. Clyde. Engine	
Diameter of Cylinder 38"	Built by
Length of stroke 36"	Pressure
Revolutions per Minute 75	Barometer Reads
Pressure of Steam in lbs. in Boiler 58	Throttle
Position of Throttle Valve Full Open	Regulator
Vacuum per Gauge in inches 22.75	REMARKS:
Temperature of Hot Well 126°	Same as H.P. Diagram
Scale of Spring 12	
Inside Diameter of Feed Pipe	
" " Exhaust Pipe	
Valves	

The average M. E. P. is 36.25# + 35# = 71.25 lbs. 71.25 ÷ 2 = 35.625 lbs.

For diagrams "d" and "e" we have no drop in receiver. The receiver pressure is 6.75 lbs. gauge or 21.75 lbs absolute. The M. E. P. of L. P. top is 15 lbs. The M. E. P. of L. P. bottom is 13.875 lbs.

The average M. E. P. is 15# + 13.875# = 28.875 lbs. 28.875 ÷ 2 = 14.875 lbs.

The constant for H. P. cylinder we found to be 0.02741.

Now for H. P. cylinder the I. H. P. is thus found to be $0.02741 \times 35.625 \times 75 \times 3 = 219.712$ I. H. P.

The constant for L. P. cylinder was 0.06873.

The I. H. P. L. P. cylinder is thus found to be $0.06873 \times 14.875 \times 75 \times 3 = 229.95$ I. H. P., say 230.

The collective I. H. P. = 219.712 + 230 = 449.712.

The aggregate equivalent M. P. referred to L. P. piston is, therefore,

$$\frac{35.625}{2.56} + 14.875 = 13.91\# + 14.875\# = 28.785 \text{ lbs.*}$$

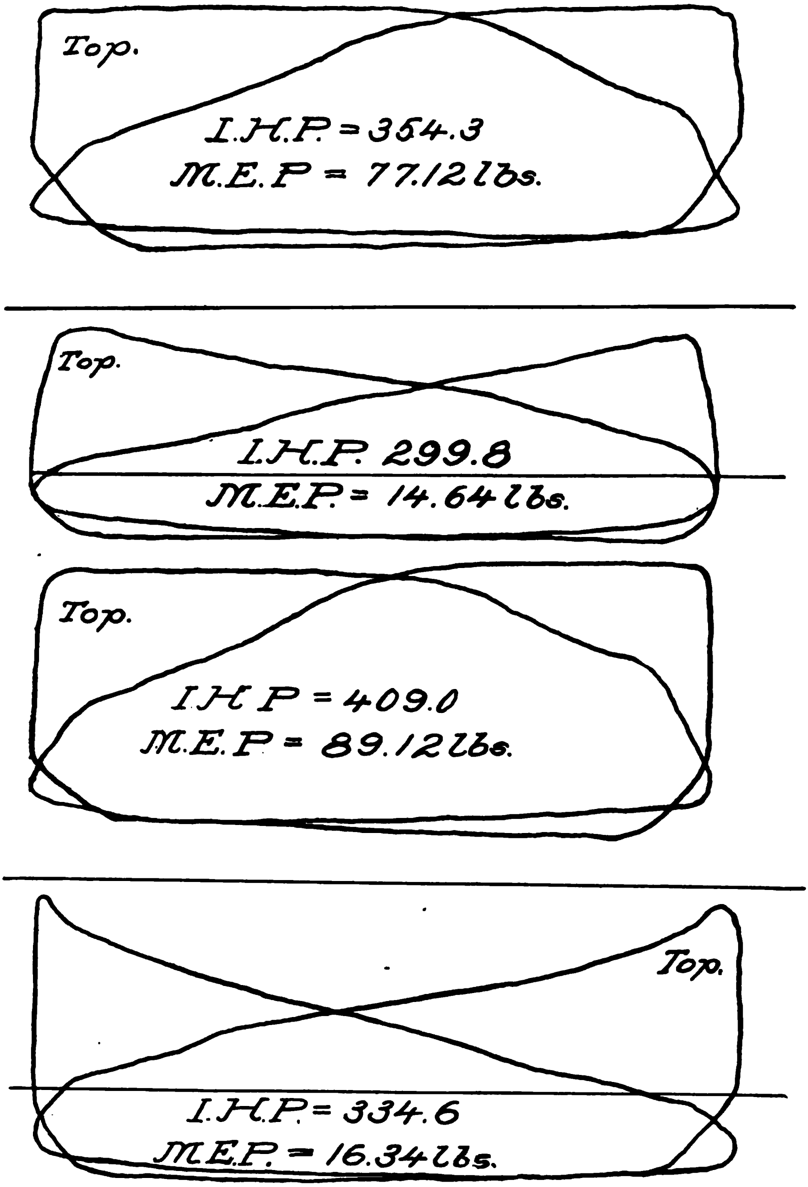
Following are two series of indicator diagrams. Taken from a double screw ferryboat, whose cylinders measure

$$\frac{18'' \times 38''}{28''} \text{ and } \frac{18'' \times 38''}{28''}$$

RUN	REV.	RUN	REV.
1 and 2	128¼	5 and 6	128½
3 and 4	130	Average of all Runs	128.9

*A close perusal of the diagrams from the G. W. Clyde will prove the uncertainty and, in fact, unreliability of ordinary indicator pipes as fitted. If on trial trip the ordinary method of one instrument to each cylinder is insisted upon, then before any data is taken, diagrams with short connections should be made, and hence a correction factor determined. After this has been done, we have a check for the diagrams, and no error need be introduced.

SERIES 1

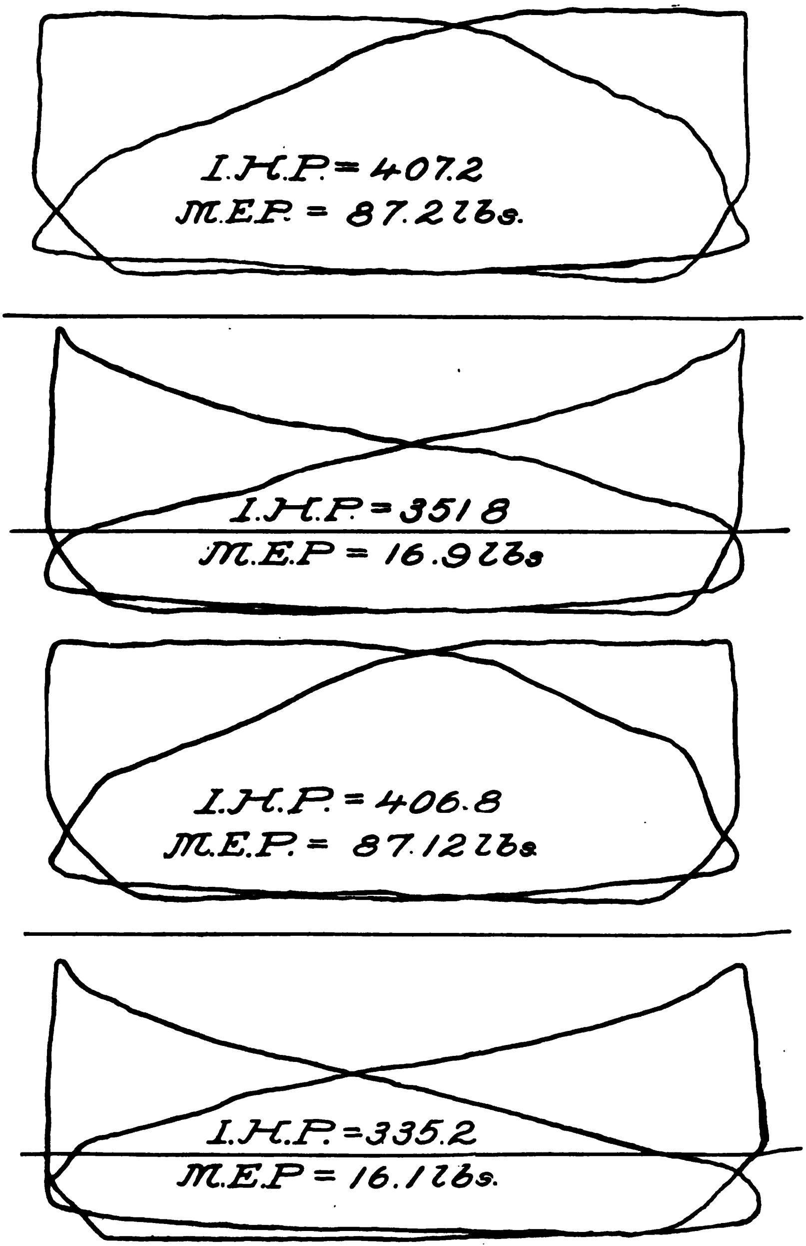


RUN No. 1A

Steam 150 For'd Rec. 24 Aft. Rec. 24 Vac. 24″ Rev. 127
I. H. P., F. H. P. 354.3 I. H. P., F. L. P. 299.8
I. H. P., A. H. P. 409.0 I. H. P., A. L. P. 334.6 Total, I. H. P. 1397.7

Throttle wide open. Gear: { F. H. P. linked in $3\frac{3}{16}$″. A. H. P. linked in $2\frac{11}{16}$″.
F. L. P. linked full out. A. L. P. linked full out.

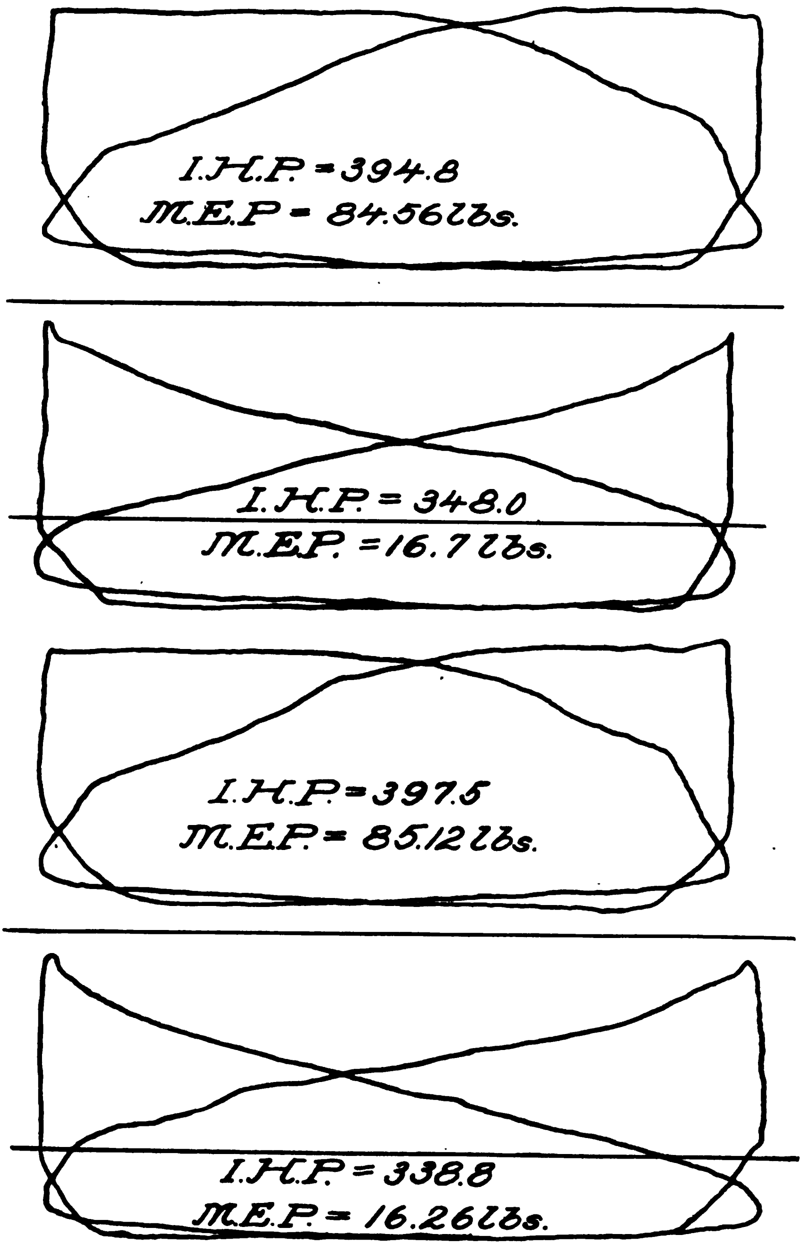
Series 1



Run No. 2A

Steam 136 For'd Rec. 21 Aft. Rec. 21 Vac. 24½″ Rev. 128
I. H. P., F. H. P. 407.2 I. H. P., F. L. P., 351.8
I. H. P., A. H. P. 406.8 I. H. P., A. L. P., 335.2 Total, I. H. P. 1501.0
Throttle wide open. Gear same as Run "1A."

Series 1



Run No. 3A

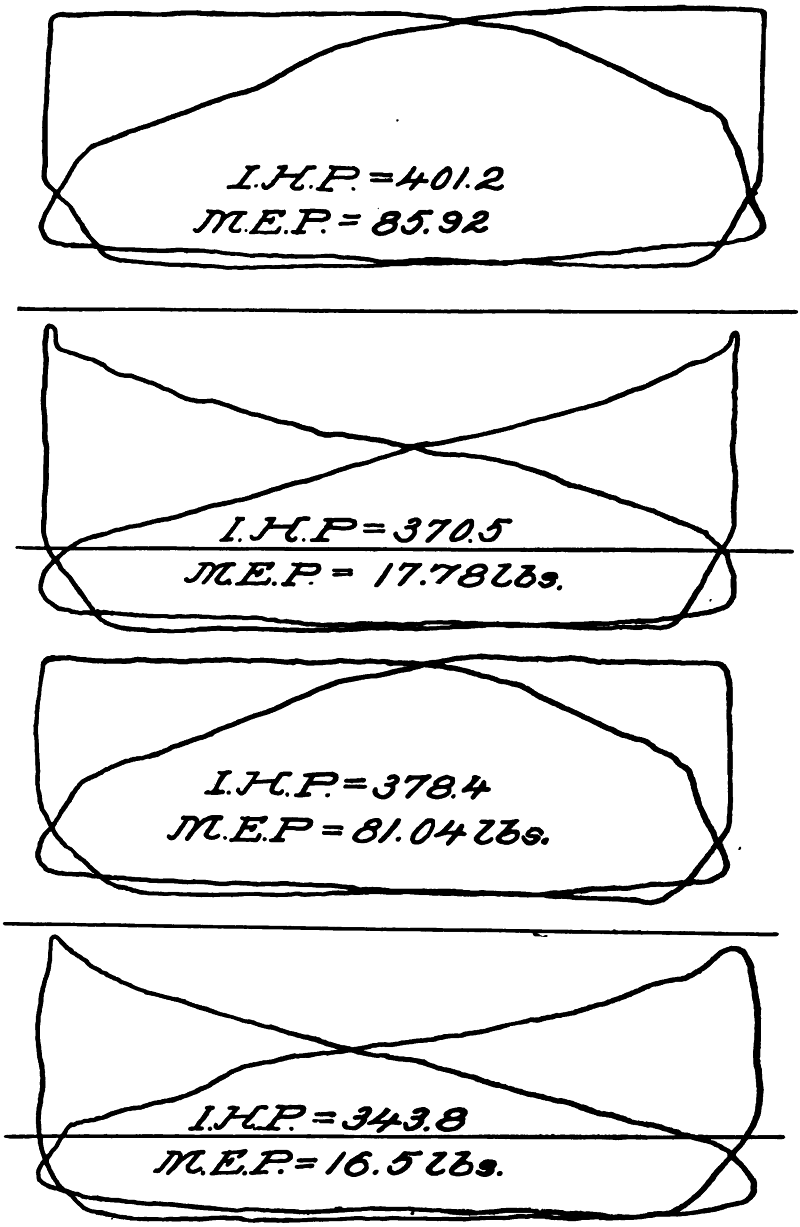
Steam 130 For'd Rec. 21 Aft. Rec. 21 Vac. $24\frac{1}{2}''$ Rev. 130

I. H. P., F. H. P. 394.8 I. H. P., F. L. P. 348.0

I. H. P., A. H. P. 397.5 I. H. P., A. L. P. 338.8 Total, I. H. P. 1479.1

Throttle wide open. Gear same as Run "1A."

Series 1



Run No. 4A

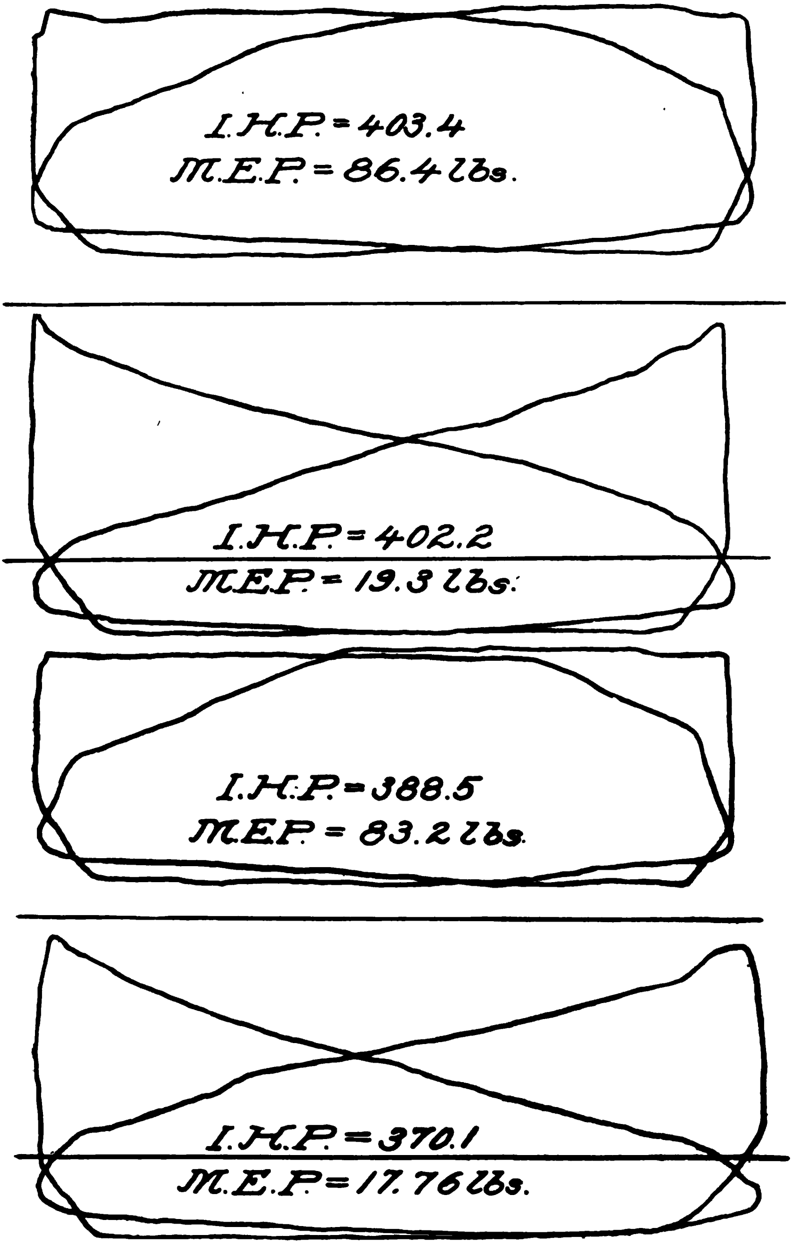
Steam 126 For'd Rec. 22 Aft. Rec. 22 Vac. 25″ Rev. 130

I. H. P., F. H. P. 401.2 I. H. P., F. L. P. 370.5

I. H. P., A. H. P. 378.4 I. H. P., A. L. P. 343.8 Total, I. H. P. 1493.9

Throttle wide open. Gear { F. H. P. linked in $1\frac{7}{8}''$. A. H. P. linked in $1\frac{7}{8}''$.
F. L. P. linked full out. A. L. P. linked full out. }

SERIES 1



RUN No. 5A

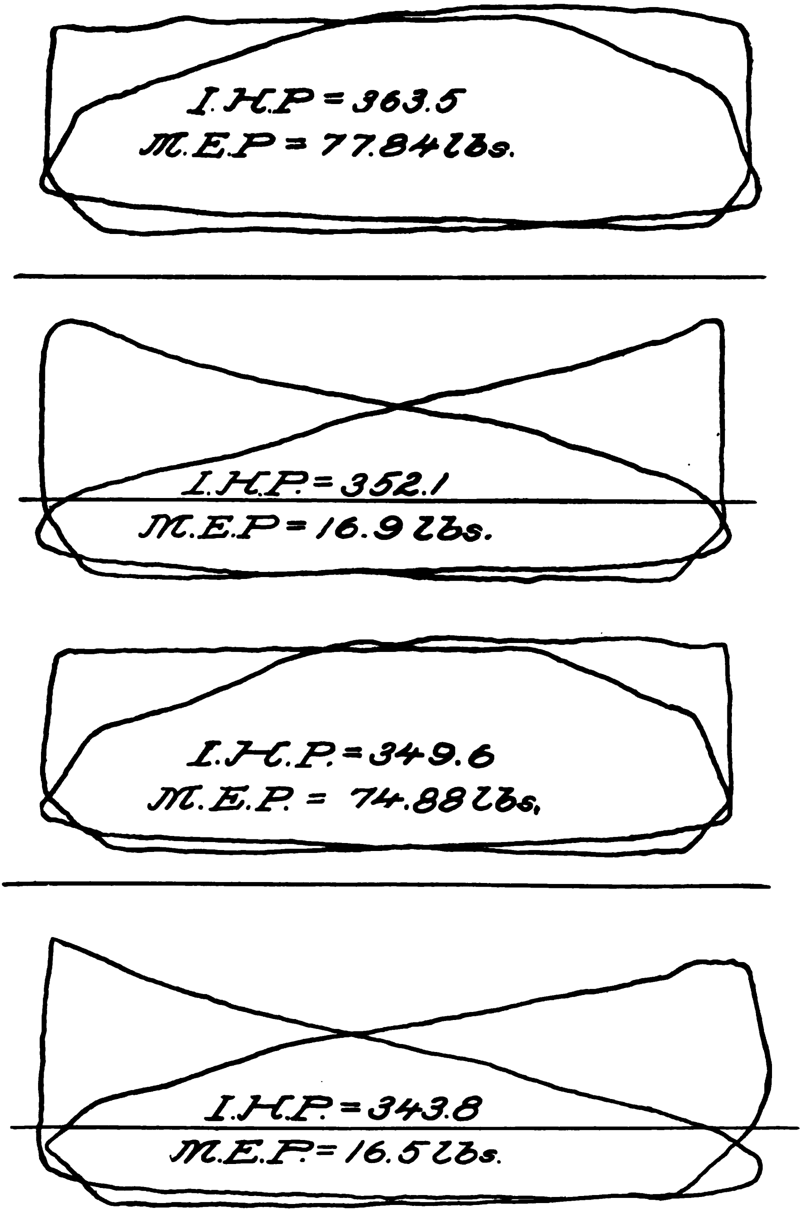
Steam 127 For'd Rec. 25 Aft. Rec. 23 Vac. 23¼″ Rev. 128

I. H. P., F. H. P. 403.4 I. H. P., F. L. P. 402.2

I. H. P., A. H. P. 388.5 I. H. P., A. L. P. 370.1 Total, I. H. P. 1564.2

Throttle wide open. Gear full out on all.

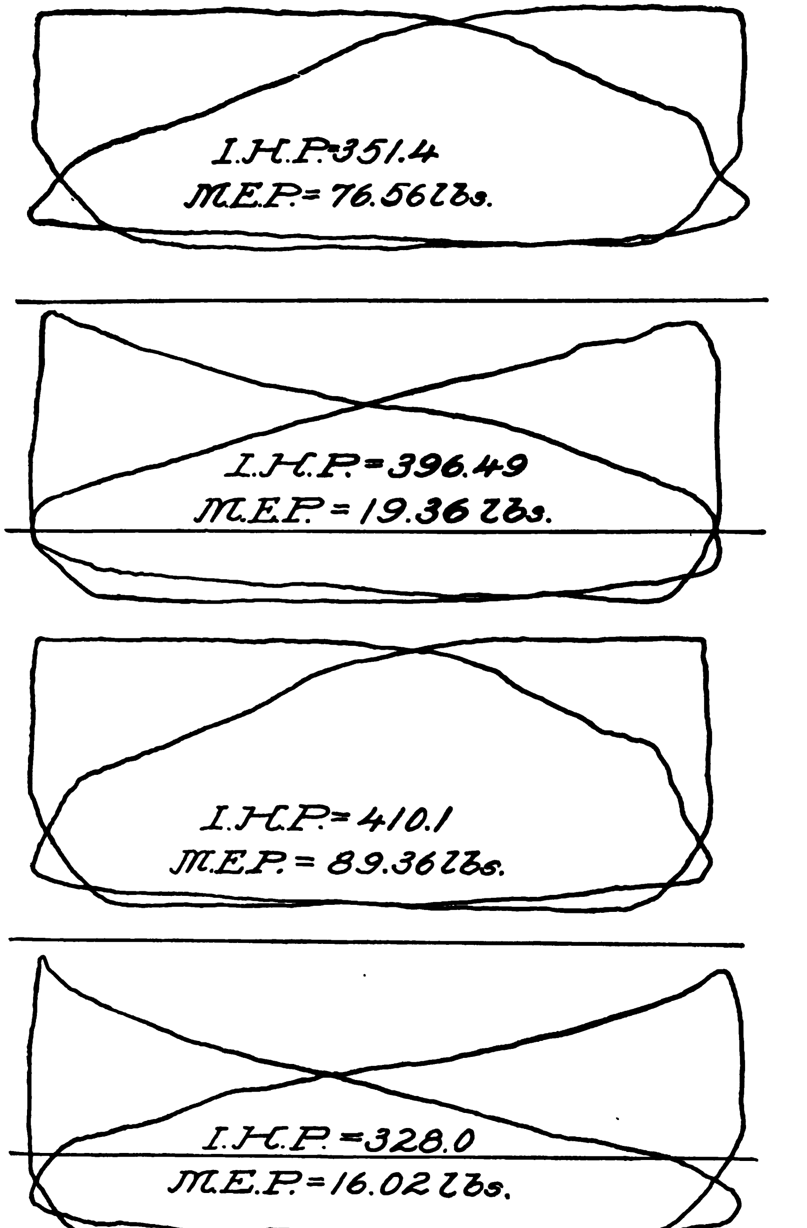
Series 1



Run No. 6A

Steam 119 For'd Rec. 21 Aft. Rec. 21 Vac. 24½″ Rev. 124
I. H. P., F. H. P. 363.5 I. H. P., F. L. P. 352.1
I. H. P., A. H. P. 349.6 I. H. P., A. L. P. 343.8 Total, I. H. P. 1409.0
Throttle wide open. Gear full out on all.

SERIES 1



RUN No. 1B

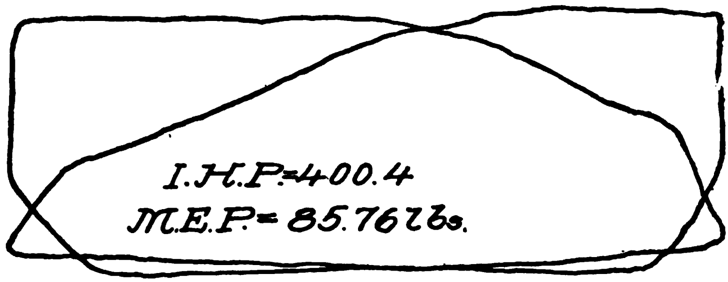
Steam 120 For'd Rec. 28 Aft. Rec. 20 Vac. 24½″ Rev. 128

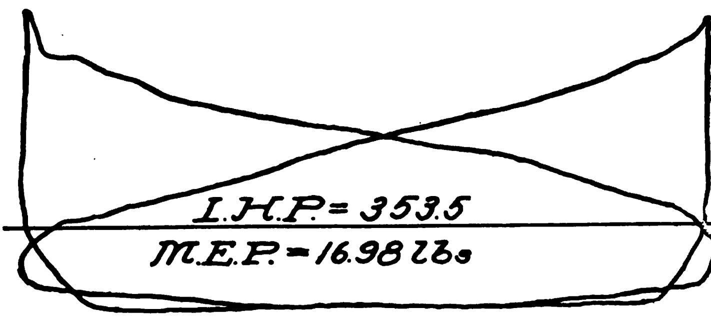
I. H. P., F. H. P. 351.4 I. H. P., F. L. P. 396.4

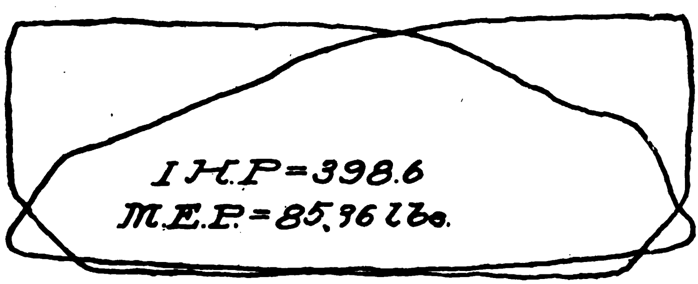
I. H. P., A. H. P. 410.1 I. H. P., A. L. P. 328.0 Total, I. H. P. 1485.9

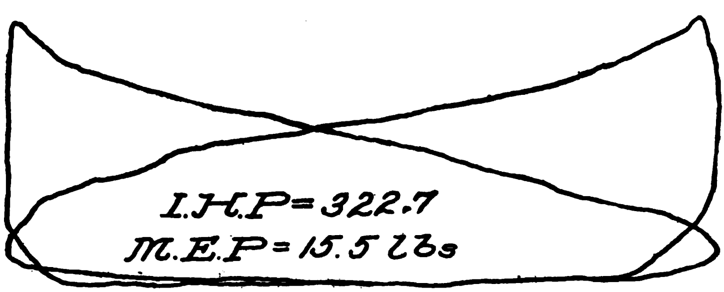
Throttle wide open. Gear same as Run "1A."

SERIES 1





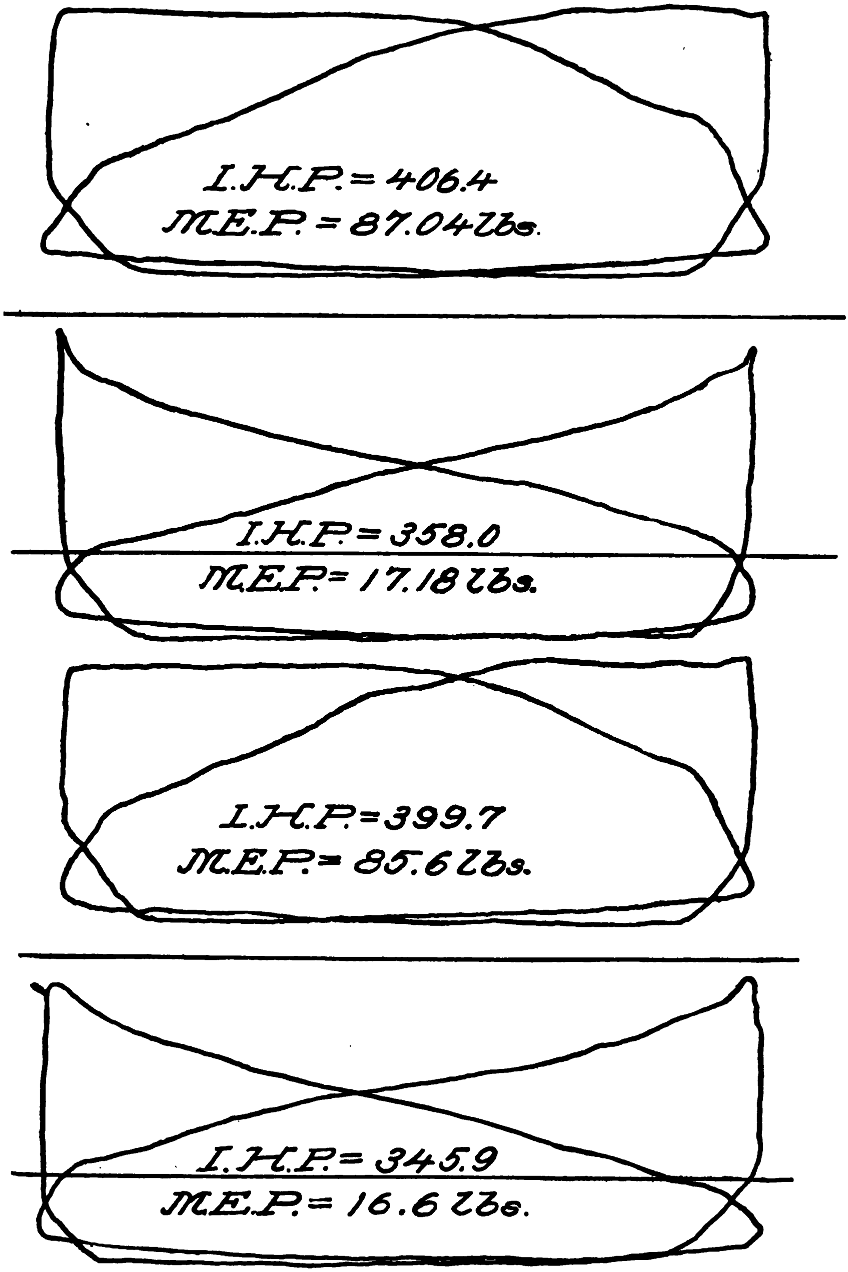




RUN No. 2B

Steam 136 For'd Rec. 21 Aft. Rec. 21 Vac. 24½″ Rev. 130
I. H. P., F. H. P. 400.4 I. H. P., F. L. P. 353.5
I. H. P., A. H. P. 398.6 I. H. P., A. L. P. 322.7 Total, I. H. P. 1475.2
Throttle wide open. Gear same as Run "1A."

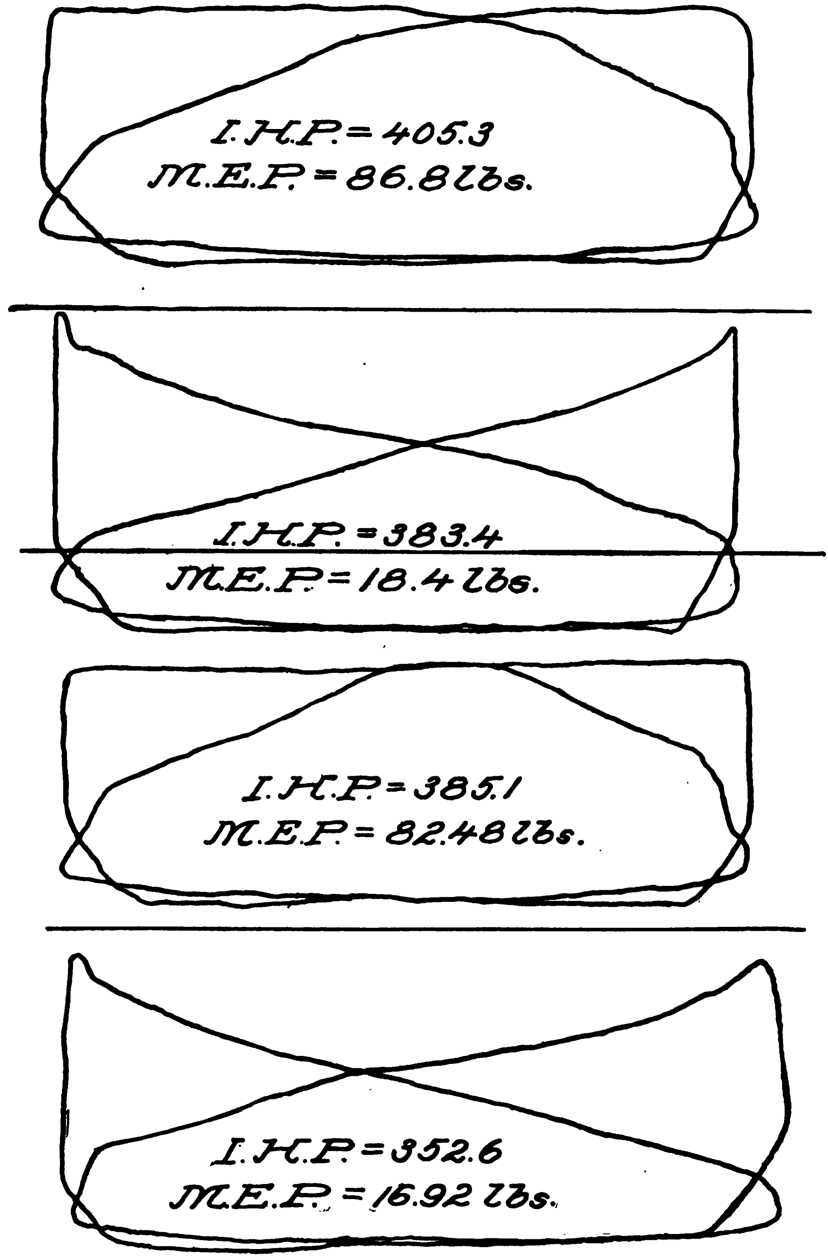
SERIES 1



RUN No. 3B

Steam 142 For'd Rec. 24 Aft. Rec. 24 Vac. 24½″ Rev. 130
I. H. P., F. H. P. 406.4 I. H. P., F. L. P. 358.0
I. H. P., A. H. P. 399.7 I. H. P., A. L. P. 345.9 Total, I. H. P. 1510.0
Throttle wide open. Gear same as Run "1A."

SERIES 1



RUN No. 4B

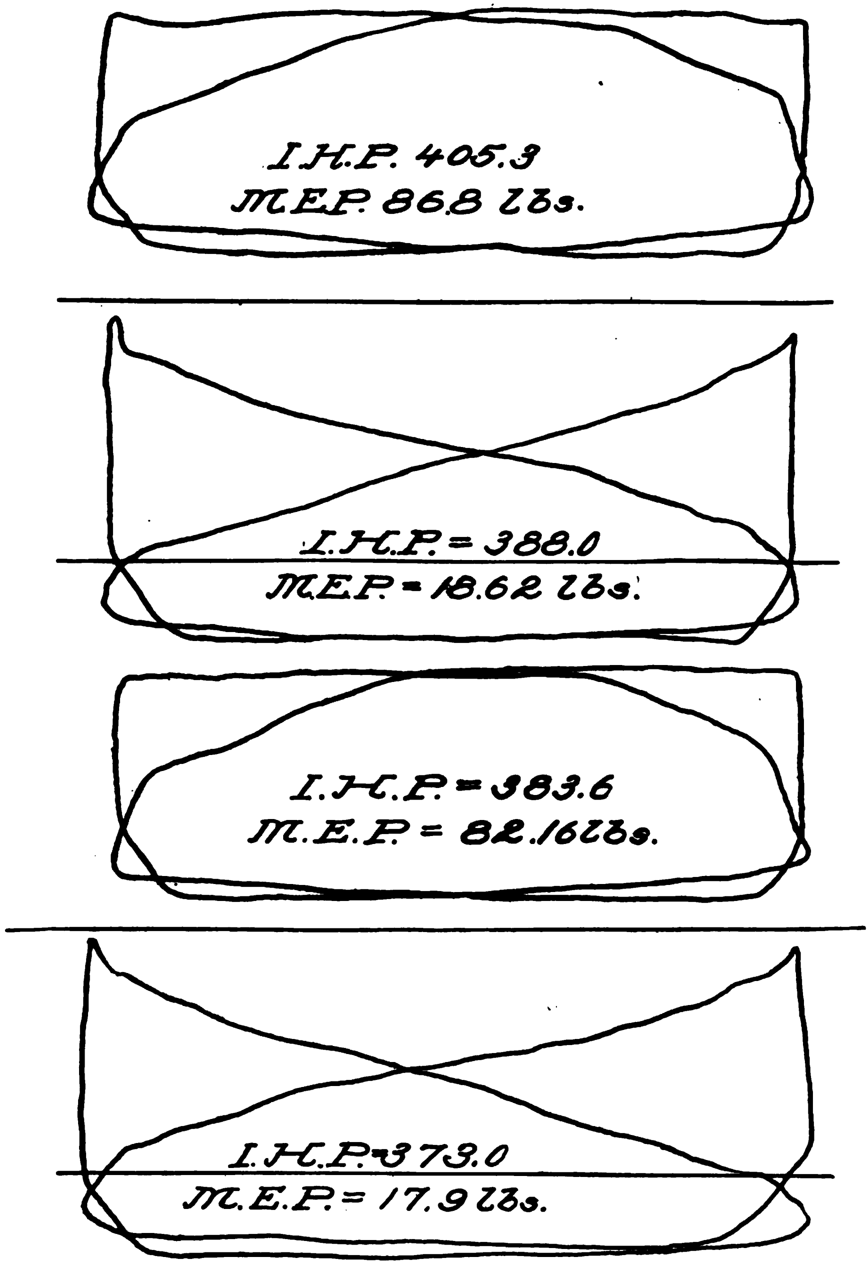
Steam 127 For'd Rec. 23 Aft. Rec. 23 Vac. 24½″ Rev. 130

I. H. P., F. H. P. 405.3 I. H. P., F. L. P. 383.4

I. H. P., A. H. P. 385.1 I. H. P., A. L. P. 352.6 Total, I. H. P. 1526.4

Throttle wide open. Gear same as Run "4A."

SERIES 1



RUN No. 5B

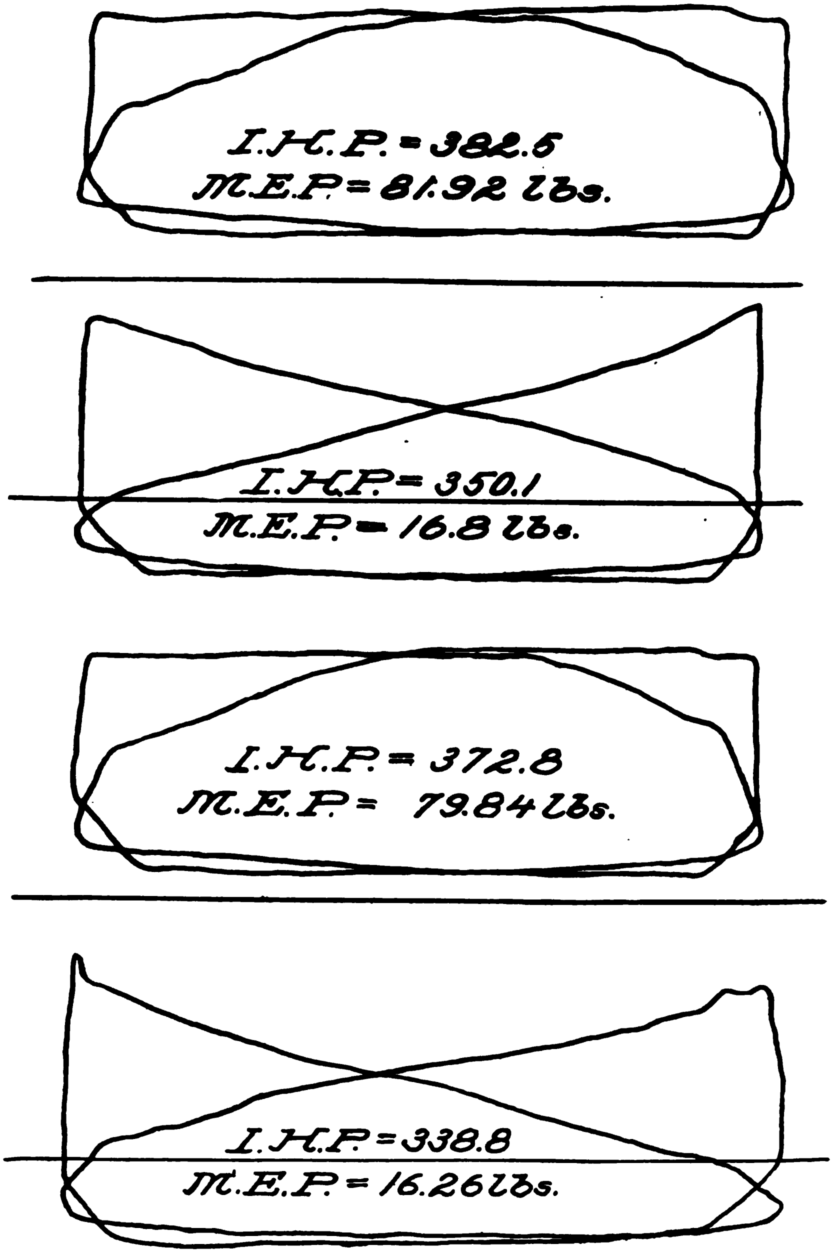
Steam 130 For'd Rec. 25 Aft. Rec. 24 Vac. 24½″ Rev. 130

I. H. P., F. H. P. 405.3 I. H. P., F. L. P. 388.0

I. H. P., A. H. P. 383.6 I. H. P., A. L. P. 373.0 Total, I. H. P. 1549.9

Throttle wide open. Gear full out on all.

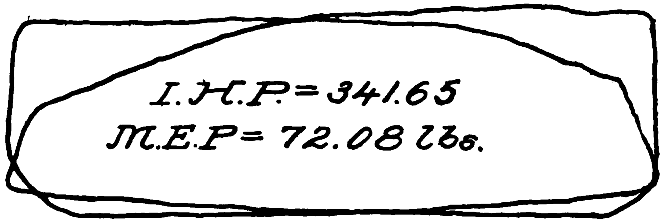
SERIES 1

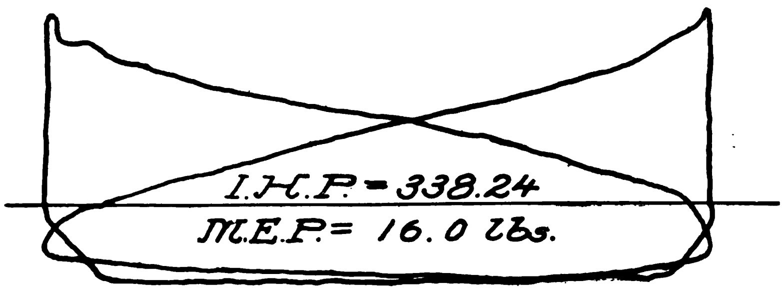


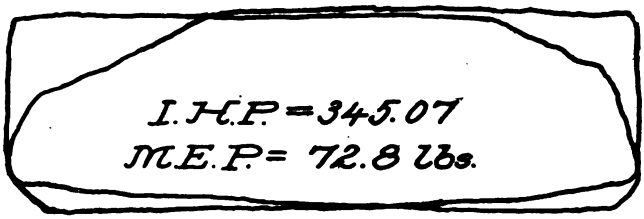
RUN No. 6B

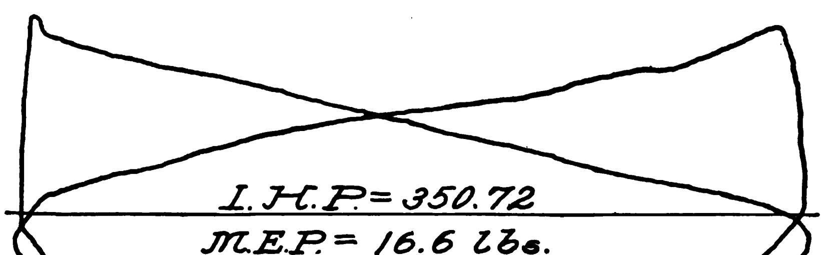
Steam 130 For'd Rec. 24 Aft. Rec. 25 Vac. 25½″ Rev. 130
I. H. P., F. H. P. 382.5 I. H. P., F. L. P. 350.1
I. H. P., A. H. P. 372.8 I. H. P., A. L. P. 338.8 Total, I. H. P. 1444.2
Throttle wide open. Gear full out on all.

SERIES 2









RUN No. 1A

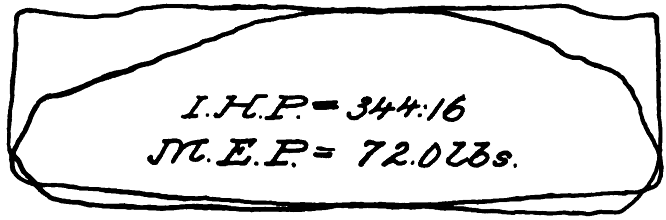
Steam 124 L. P. Rec. 23 Vac. $25\frac{1}{2}''$ Rev. 132

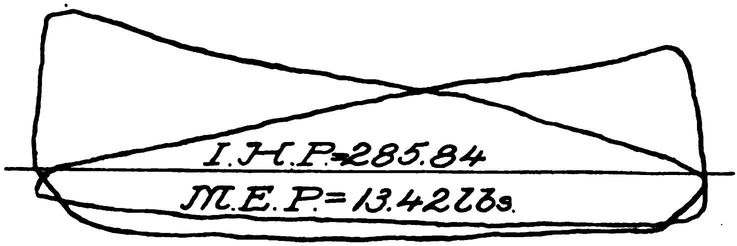
I. H. P., F. H. P. 341.65 I. H. P., F. L. P. 338.24

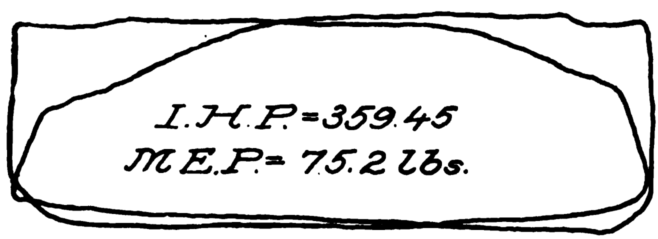
I. H. P., A. H. P. 345.07 I. H. P., A. L. P. 350.72 Total, I. H. P. 1375.88

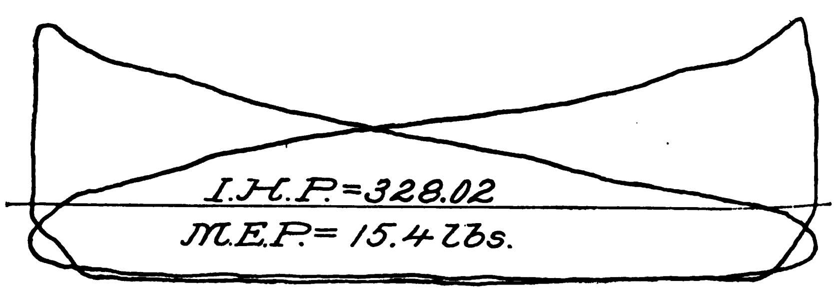
Throttle wide open. Full Gear.

Series 2









Run No. 2A

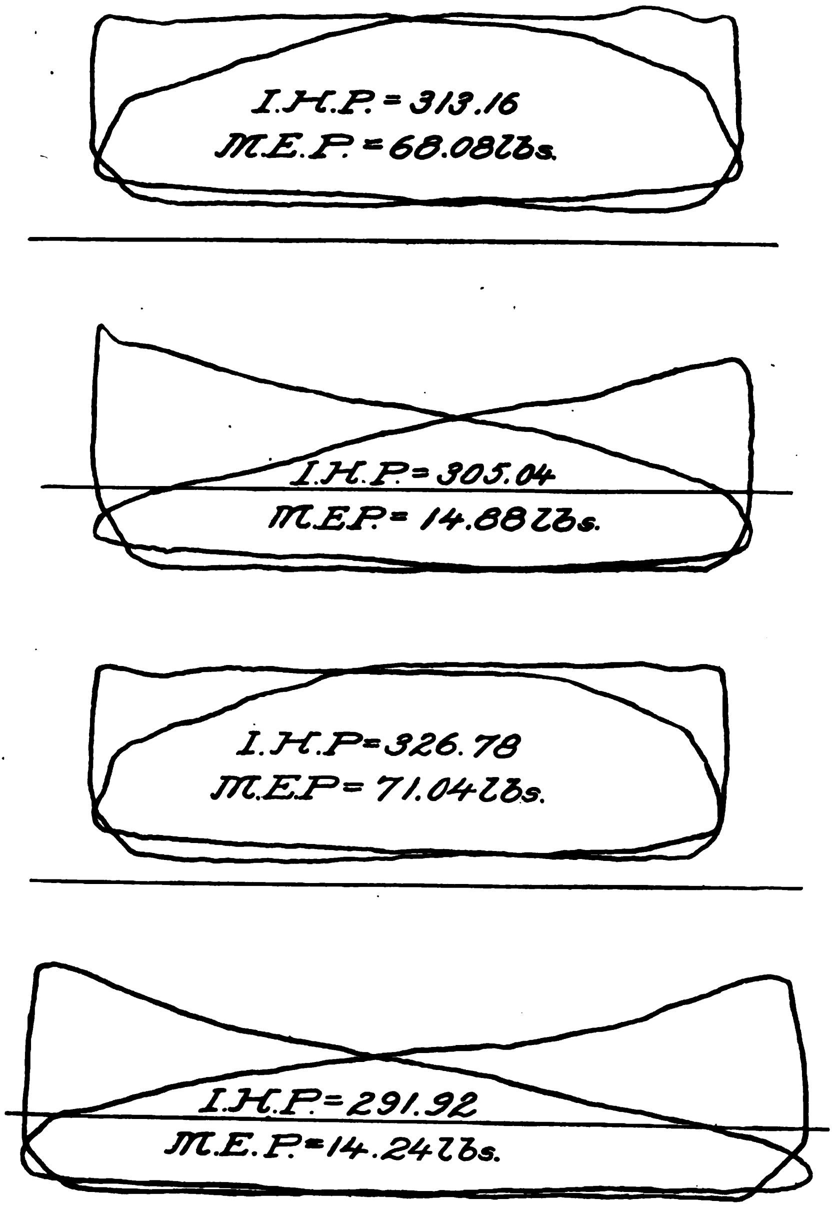
Steam 127½ L. P. Rec. 23½ Vac. 22″ Rev. 133

I. H. P., F. H. P. 344.16 I. H. P., F. L. P. 285.84

I. H. P., A. H. P. 359.45 I. H. P., A. L. P. 328.02 Total, I. H. P. 1317.47

Throttle wide open. Full Gear.

SERIES 2



RUN No. 3A

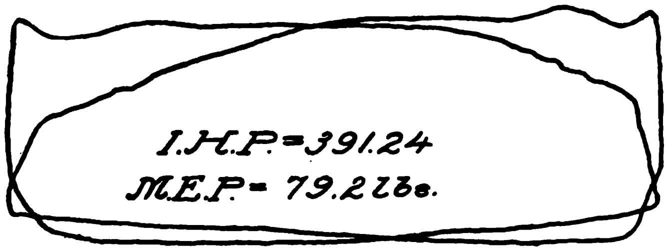
Steam 112 L. P. Rec. 18 Vac. 25″ Rev. 128

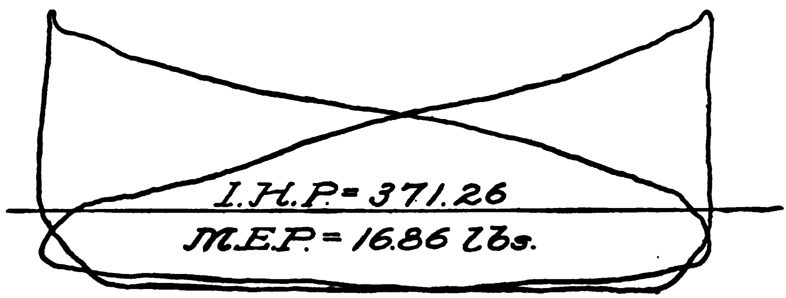
I. H. P., F. H. P. 313.16 I. H. P., F. L. P. 305.04

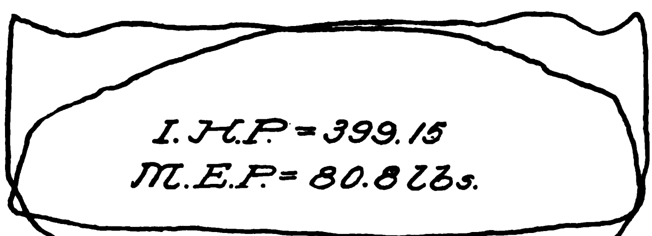
I. H. P., A. H. P. 326.78 I. H. P., A. L. P. 291.92 Total, I. H. P. 1236.90

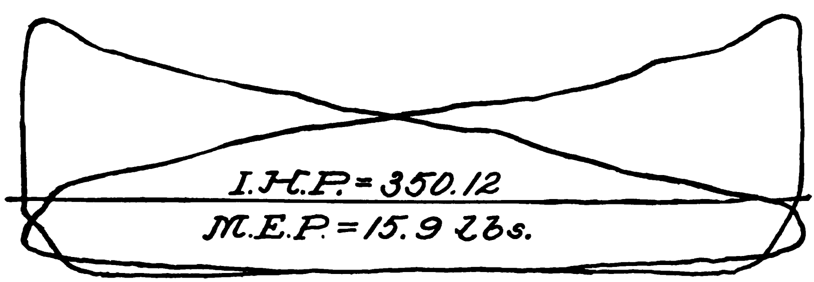
Throttle wide open. Full Gear.

Series 2









Run No. 4A

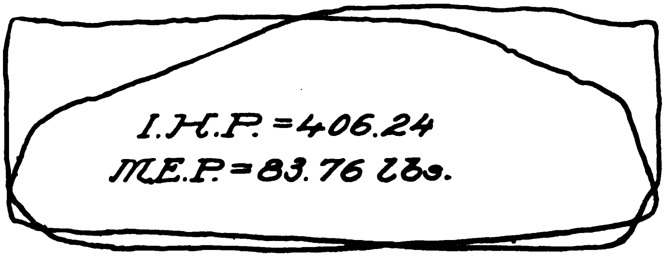
Steam 122 L. P. Rec. 19½ Vac. 25¾″ Rev. 137½

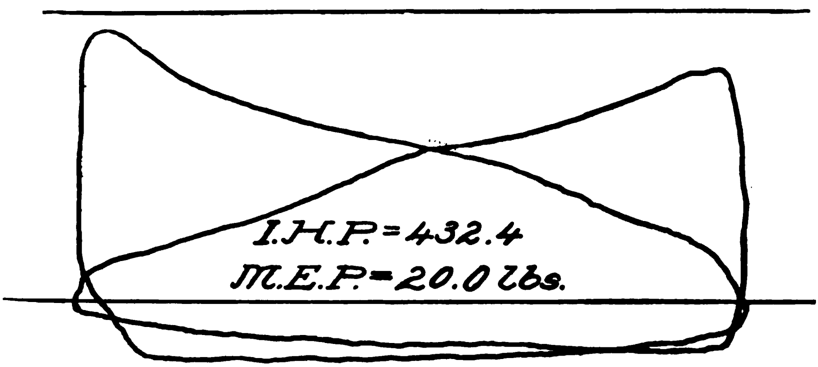
I. H. P., F. H. P. 391.24 I. H. P., F. L. P. 371.26

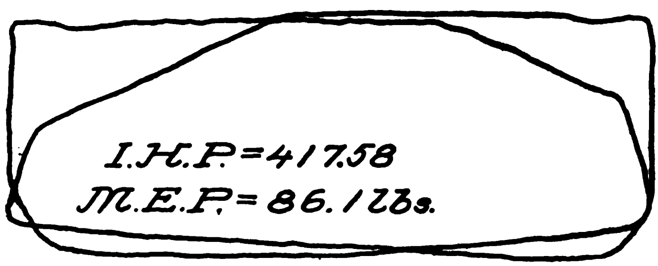
I. H. P., A. H. P. 399.15 I. H. P., A. L. P. 350.12 Total, I. H. P. 1511.77

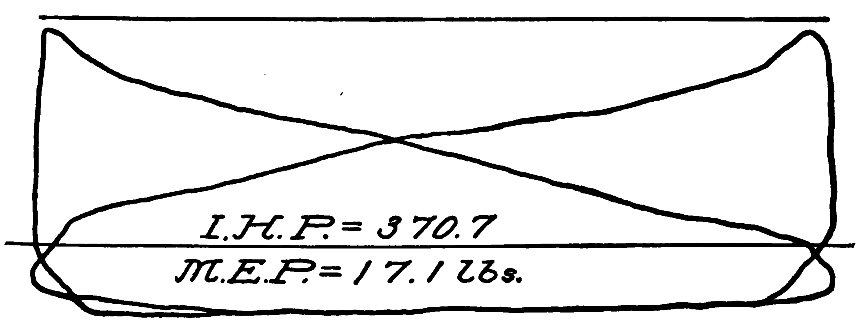
Throttle wide open. Full Gear.

SERIES 2









RUN No. 5A

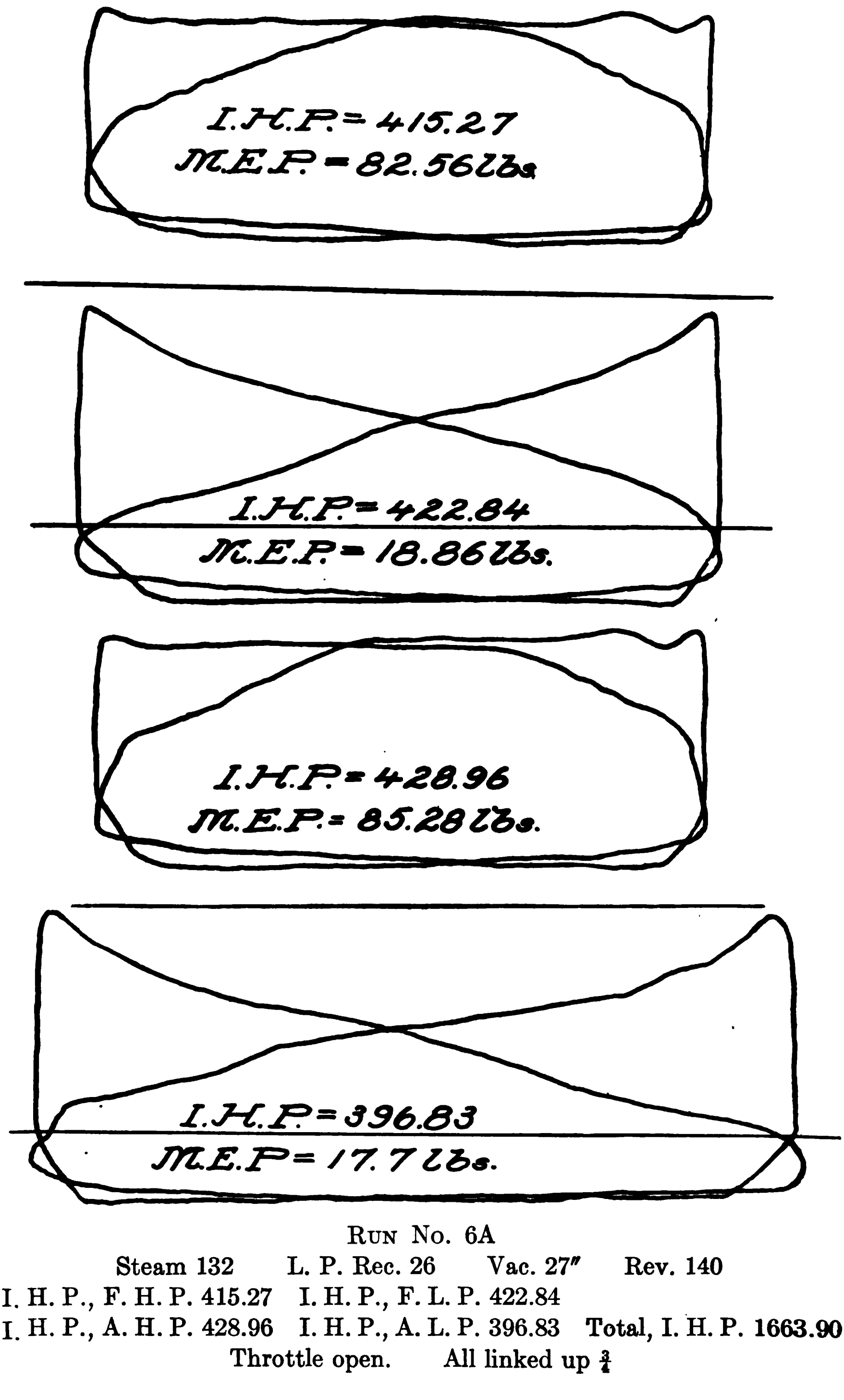
Steam 160 L. P. Rec. 28 Vac. 23½″ Rev. 135

I. H. P., F. H. P. 406.24 I. H. P., F. L. P. 432.4

I. H. P., A. H. P. 417.58 I. H. P., A. L. P. 370.7 Total, I. H. P. 1626.92

Throttle Half Open. All linked up ¾.

SERIES 2



RUN NO. 6A

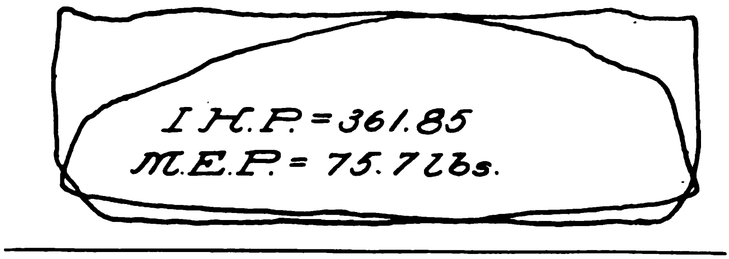
Steam 132 L. P. Rec. 26 Vac. 27″ Rev. 140

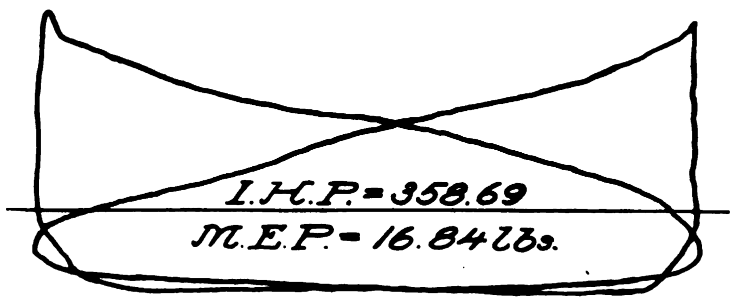
I. H. P., F. H. P. 415.27 I. H. P., F. L. P. 422.84

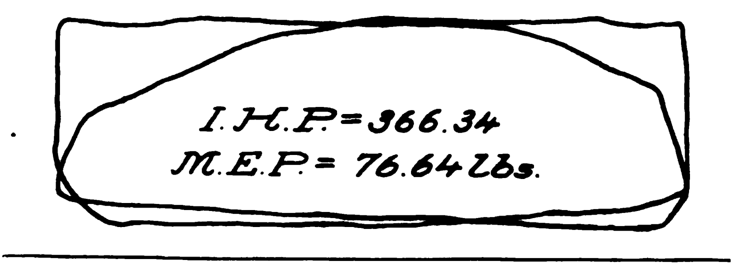
I. H. P., A. H. P. 428.96 I. H. P., A. L. P. 396.83 Total, I. H. P. 1663.90

Throttle open. All linked up ¾

SERIES 2



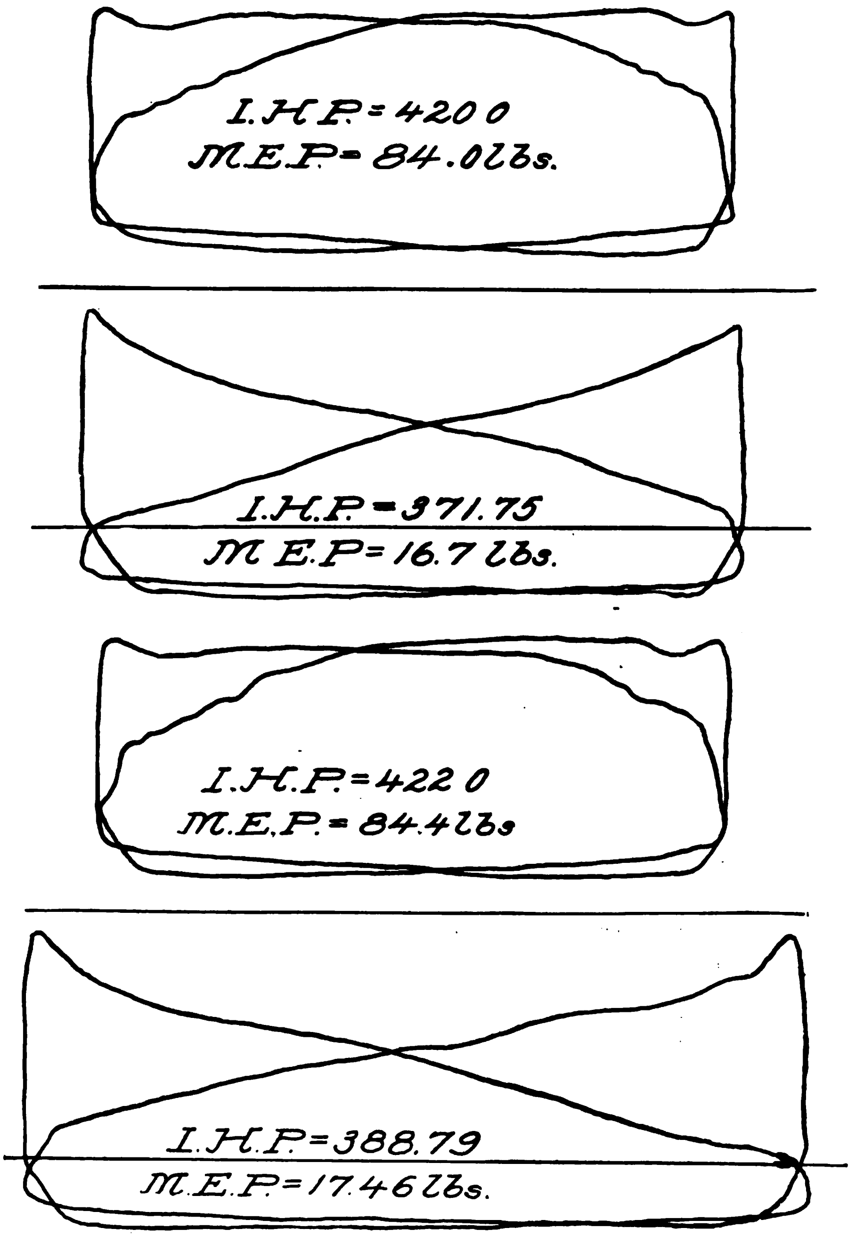




RUN No. 1B

Steam 124   L. P. Rec. $22\frac{1}{2}$ Vac. $25\frac{1}{2}''$ Rev. 133
I. H. P., F. H. P. 361.85   I. H. P., F. L. P. 358.69
I. H. P., A. H. P. 366.34   I. H. P., A. L. P.   Total, I. H. P.
Throttle wide open.   Full Gear.

SERIES 2



RUN No. 2B

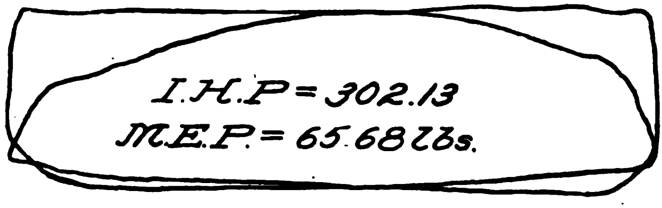
Steam 132½ L. P. Rec. 23 Vac. 24½″ Rev. 139

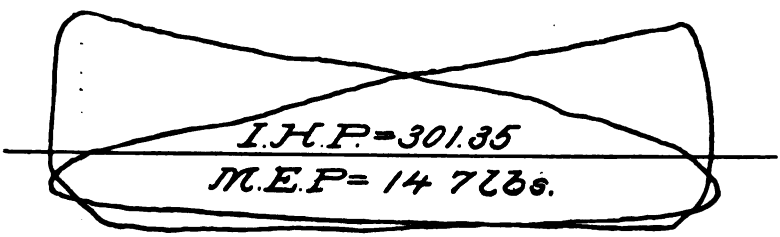
I. H. P., F. H. P. 420.0 I. H. P., F. L. P. 371.75

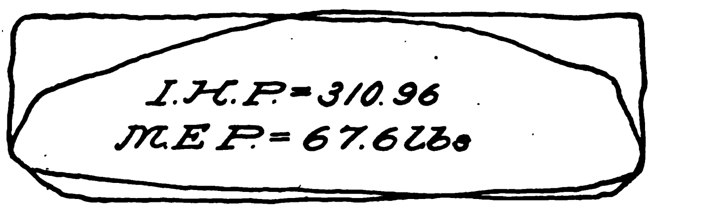
I. H. P., A. H. P. 422.0 I. H. P., A. L. P. 388.79 Total, I. H. P. 1602.54

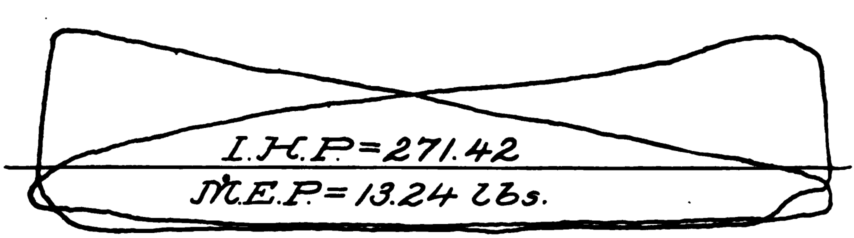
Throttle wide open. Full Gear.

SERIES 2









RUN No. 3B

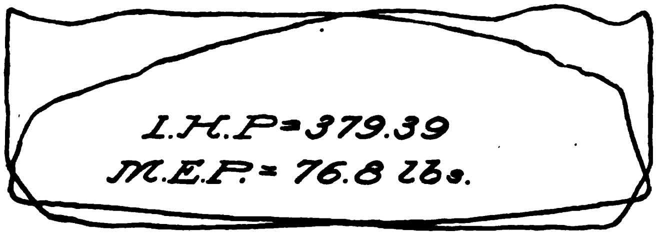
Steam 107 L. P. Rec. 18 Vac. 18″ Rev. 128

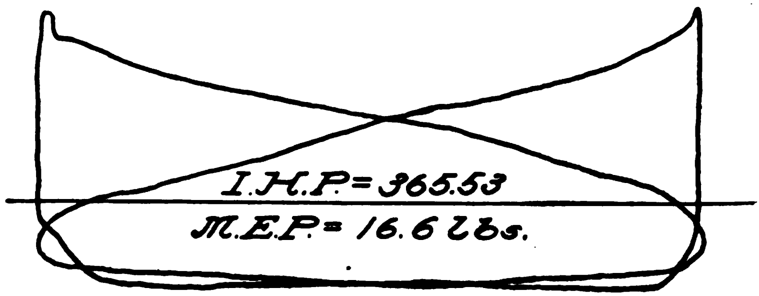
I. H. P., F. H. P. 302.13 I. H. P., F. L. P. 301.35

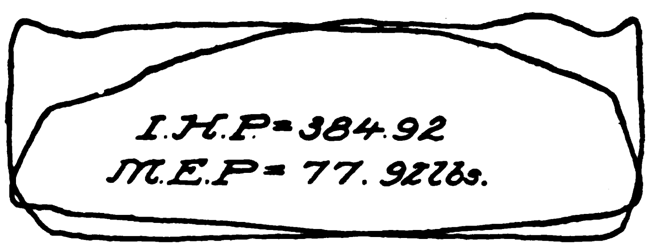
I. H. P., A. H. P. 310.96 I. H. P., A. L. P. 271.42 Total, I. H. P. 1185.86

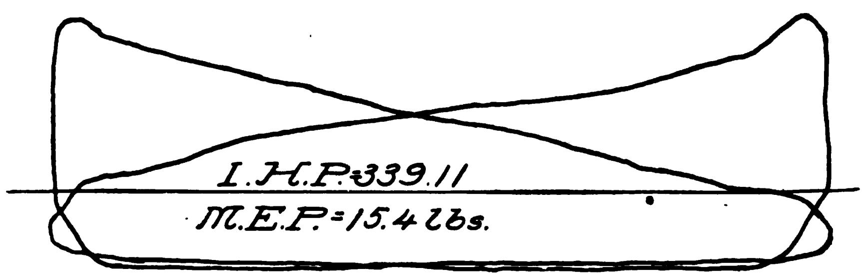
Throttle wide open. Full Gear.

SERIES 2









RUN No. 4B

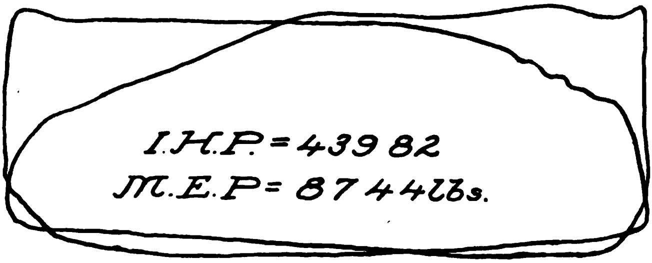
Steam 118 L. P. Rec. 18½ Vac. 26¾″ Rev. 137½

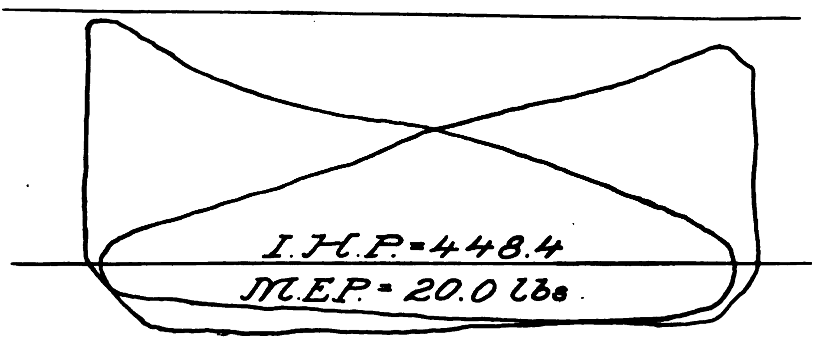
I. H. P., F. H. P. 379.39 I. H. P., F. L. P. 365.53

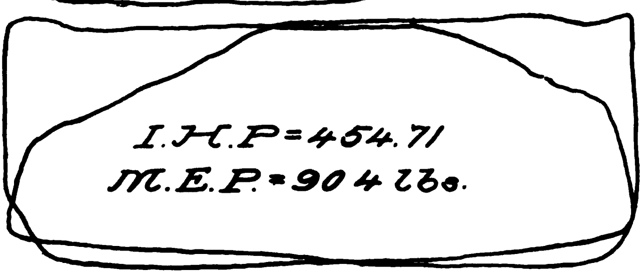
I. H. P., A. H. P. 384.92 I. H. P., A. L. P. 339.11 Total, I. H. P. 1468.95

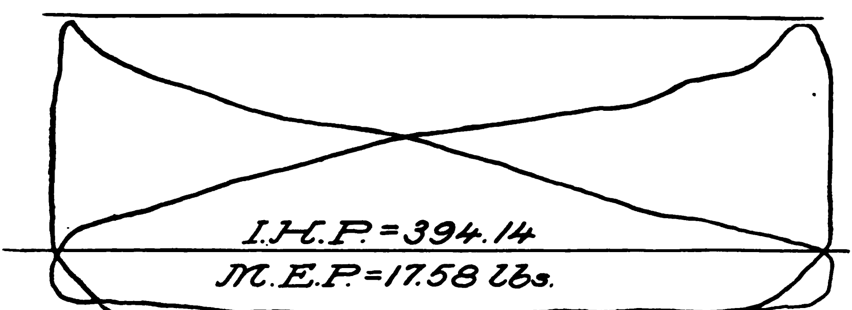
Throttle wide open. Full Gear.

SERIES 2









RUN No. 5B

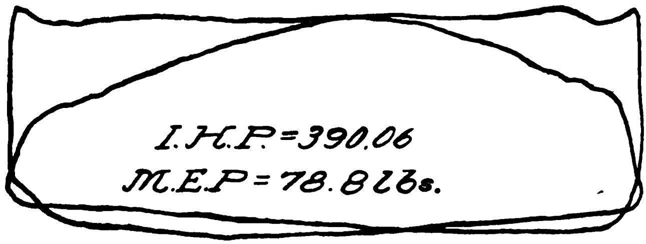
Steam 150 L. P. Rec. 28 Vac. 24″ Rev. 140

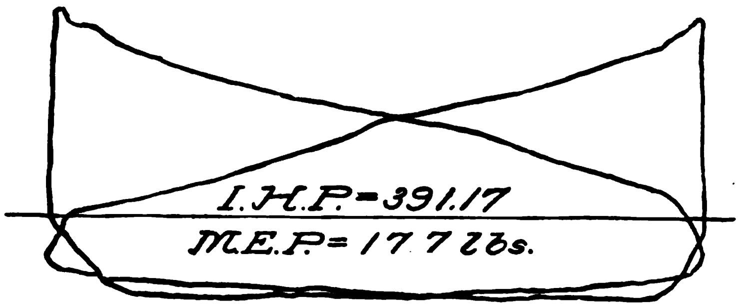
I. H. P., F. H. P. 439.82 I. H. P., F. L. P. 448.4

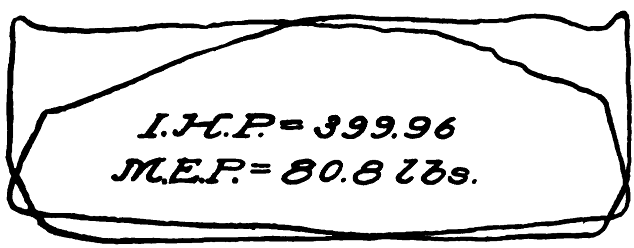
I. H. P., A. H. P. 454.71 I. H. P., A. L. P. 394.14 Total, I. H. P. 1737.07.

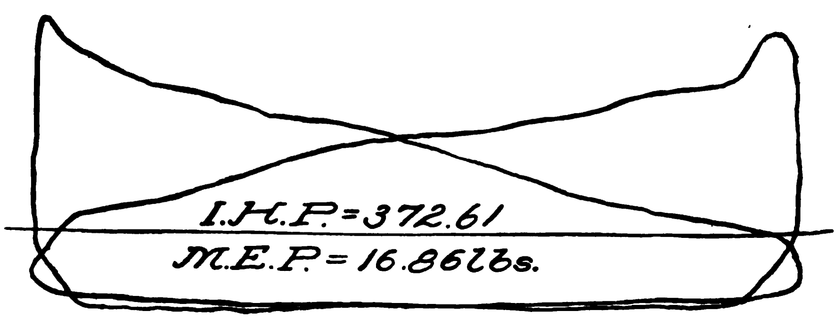
Throttle wide open. All linked up $\frac{3}{4}$.

Series 2









Run No. 6B

Steam 125 L. P. Rec. 23 Vac. 27″ Rev. 138½

I. H. P., F. H. P. 390.06 I. H. P., F. L. P. 391.17

I. H. P., A. H. P. 399.96 I. H. P., A. L. P. 372.61 Total, I. H. P. 1553.80

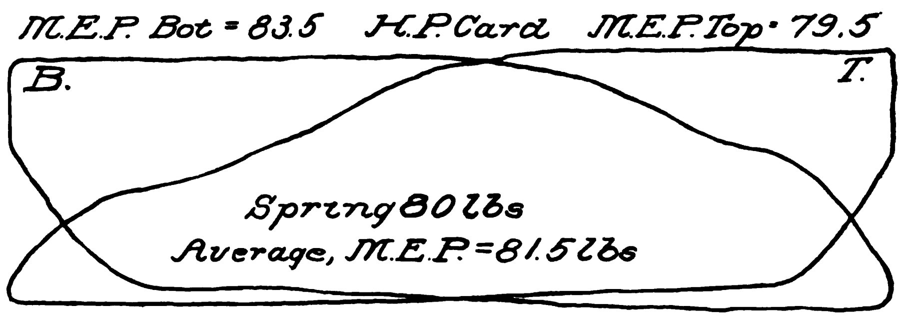
Throttle wide open. All linked up ¾.

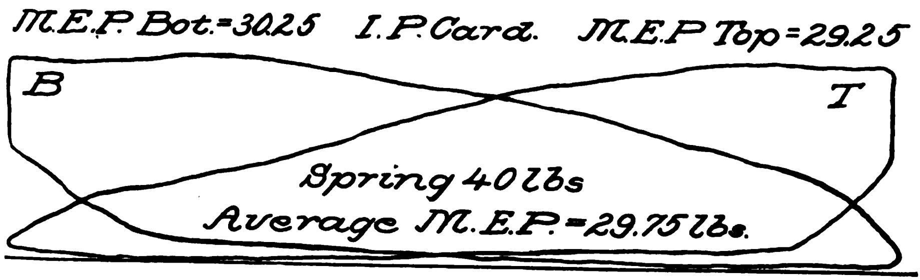
SERIES 3

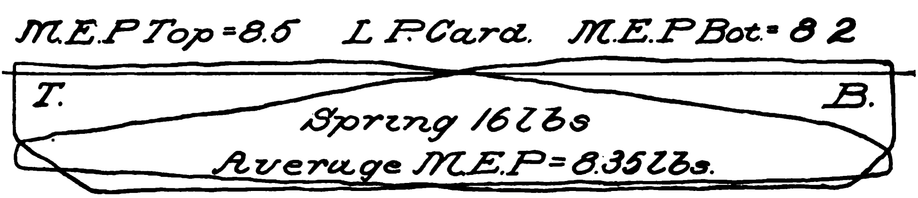
ENGINE $\dfrac{25'' \times 41\frac{1}{2}'' \times 68''}{42''}$

Steam Pressure, 170 lbs. per square inch designed. Boiler Pressure, 150 lbs. on trial. 1st Receiver 44 lbs. 2d Receiver $6\frac{1}{2}$ lbs. Vacuum, 26 inches.

Revolutions 86 lbs.







I. H. P. 710.68 H. P. Cyl. I. H. P. 551.10 L. P. Cyl.
I. H. P. 727.09 I. P. Cyl. Total, 1,988.87

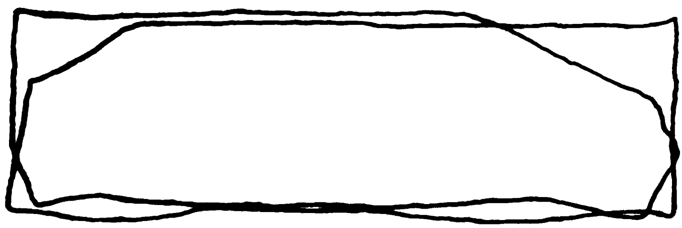
Mean Pressure, Ref. D. to L. P. Cyl. = 28.62 lbs.

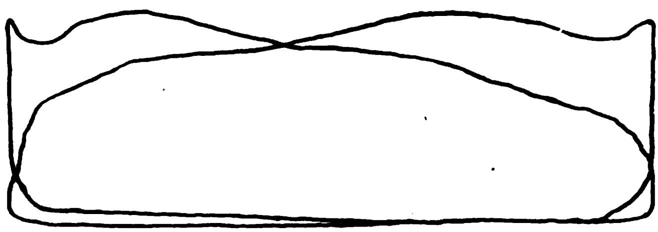
Throttle full open. All valves linked up to cut-off $29\frac{3}{4}''$ top, $26\frac{1}{2}''$ bot.

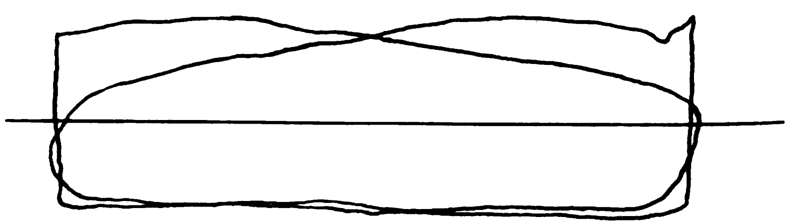
Series 4

ENGINE $\frac{34'' \times 57'' \times 104''}{63''}$

HIGH PRESSURE	INTERMEDIATE PRESSURE	LOW PRESSURE
Diam. Cylinder....34″	Diam. Cylinder....57″	Diam. Cylinder...104″
Diam. Piston Rod...9″	Diam. Piston Rod...9″	Diam. Piston Rod...9″
Stroke............63″	Stroke ....63″	Stroke............63″
Scale of Spring ...120	Scale of Spring.....60	Scale of Spring.10&20
I. H. P. Constant .2787	I. H. P. Constant..8019	I.H.P. Constant 2.6928

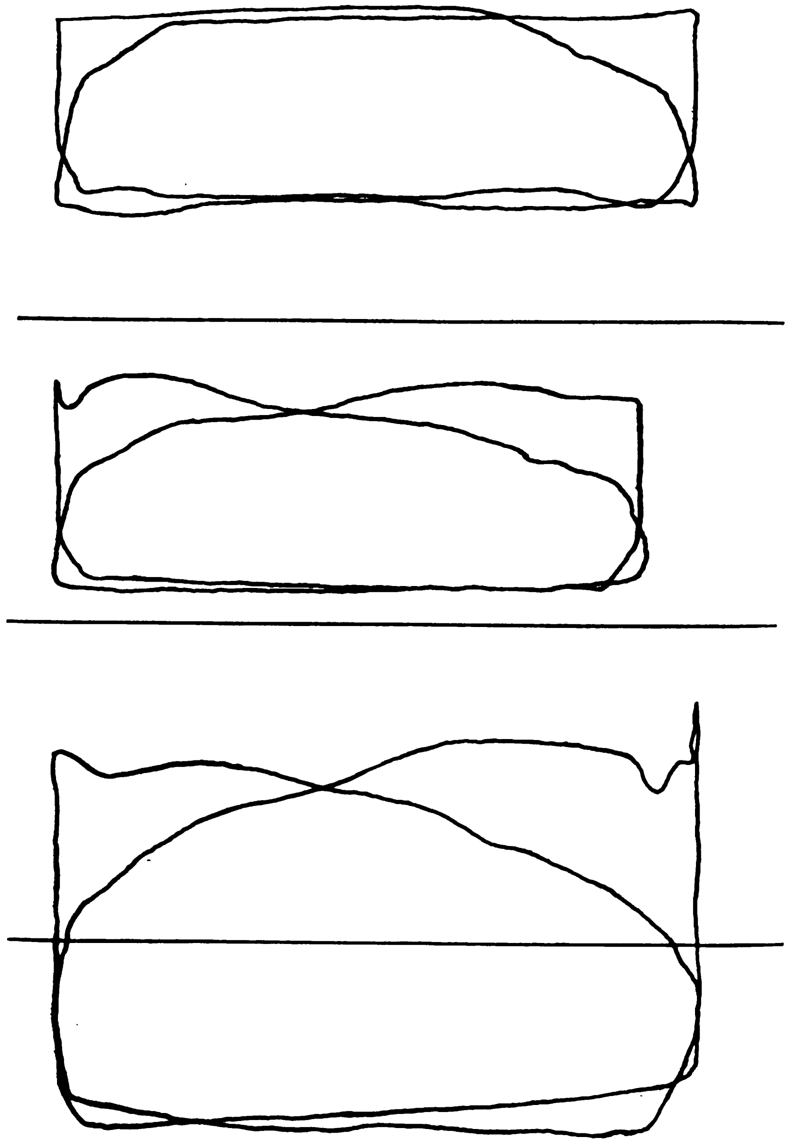






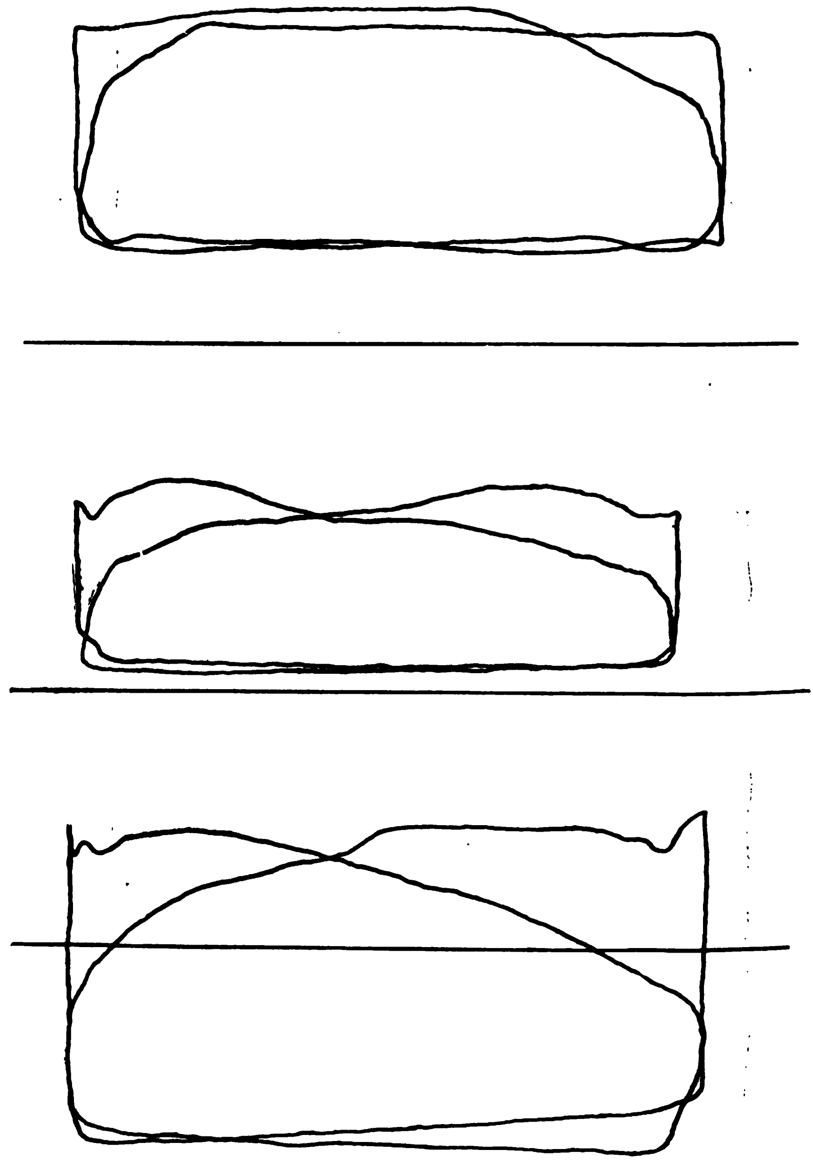
M. E. P.	I. H. P.		
H = 115.05	H = 2,661.35	Steam..............	232.
M = 52.95	M = 3,524.23	M. P. Rec.	81.
L = 17.07	L = 3,815.19	L. P. Rec...........	19.
		Vacuum	25.5″
	Total, 10,000.77	R. P. M.............	83.
		Piston Speed	871.5
		Cut Off	Full

SERIES 4



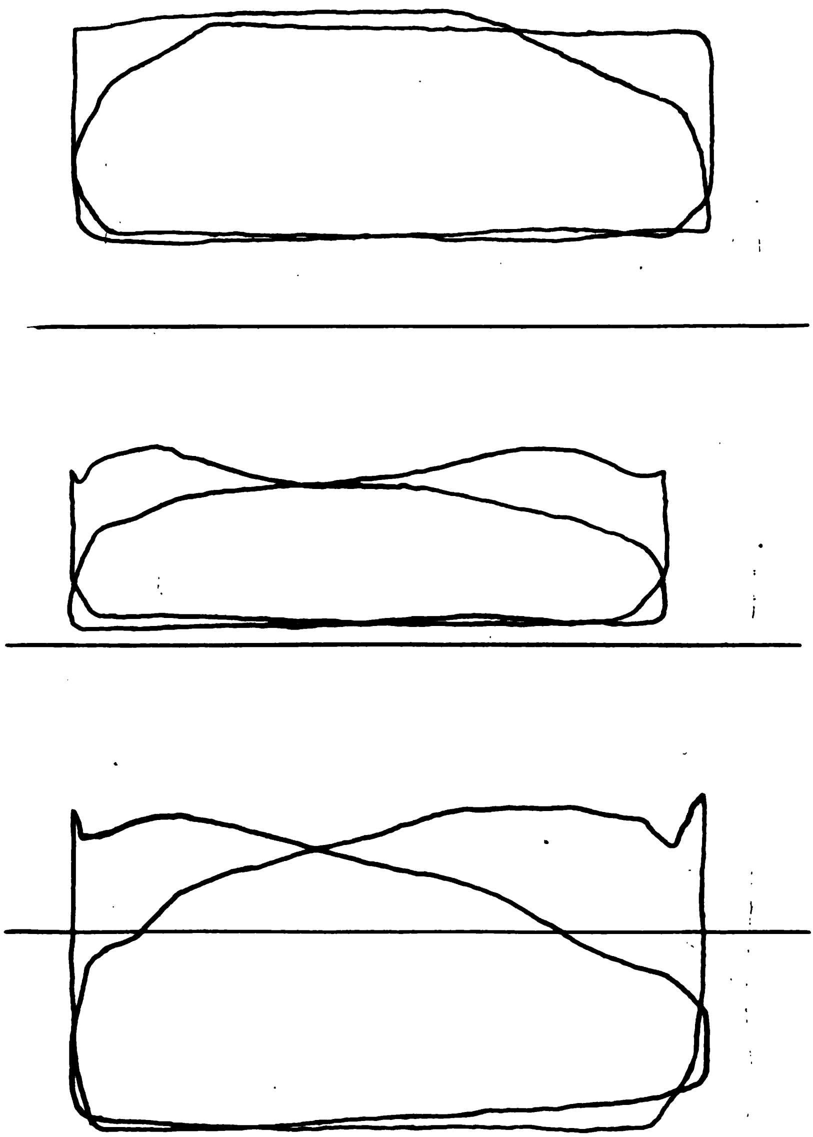
M. E. P.	I. H. P.		
H = 110.4	H = 2,524.	Steam	230.
M = 54.3	M = 3,570.	M. P. Rec.	79.
L = 16.43	L = 3,630.	L. P. Rec.	16.
	Total, 9,724.	Vacuum	24.5″
		R. P. M.	82.
		Piston Speed	861.
		Cut off	Full

SERIES 4



M. E. P.	I. H. P.		
H = 130.99	H = 2,885.	Steam	232.
M = 48.15	M = 3,050.	M. P. Rec.	65.
L = 13.71	L = 2,917.	L. P. Rec.	12.5
	Total, 8,852.	Vacuum	25.″
		R. P. M.	79.
		Piston Speed	829.5
		Cut off	Full

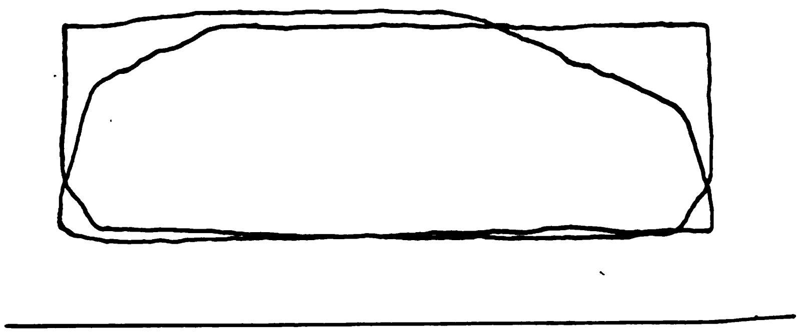
SERIES 4

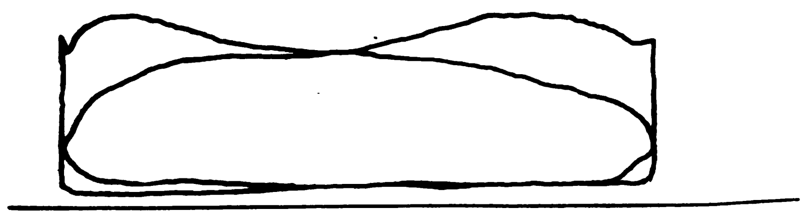


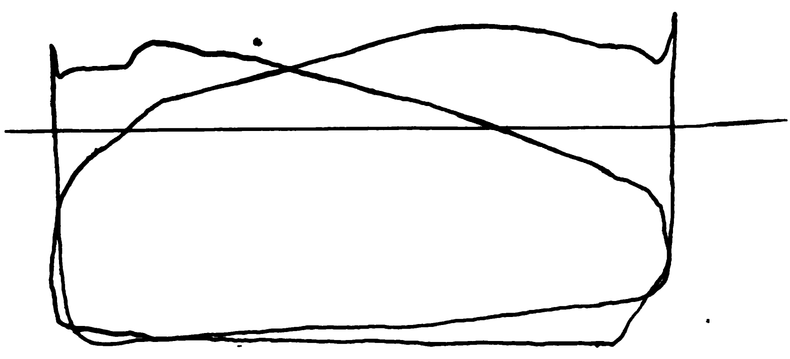
M. E. P.	I. H. P.		
H = 126.15	H = 2,707.4	Steam	225.
M = 41.55	M = 2,565.3	M. P. Rec.	62.
L = 13.50	L = 2,799.1	L. P. Rec.	11.25
		Vacuum	24.″
	Total, 8,071.8	R. P. M.	77
		Piston Speed	808.5

Cut off: H. P. = .71, M. P. = .732, L. P. = Full

Series 4



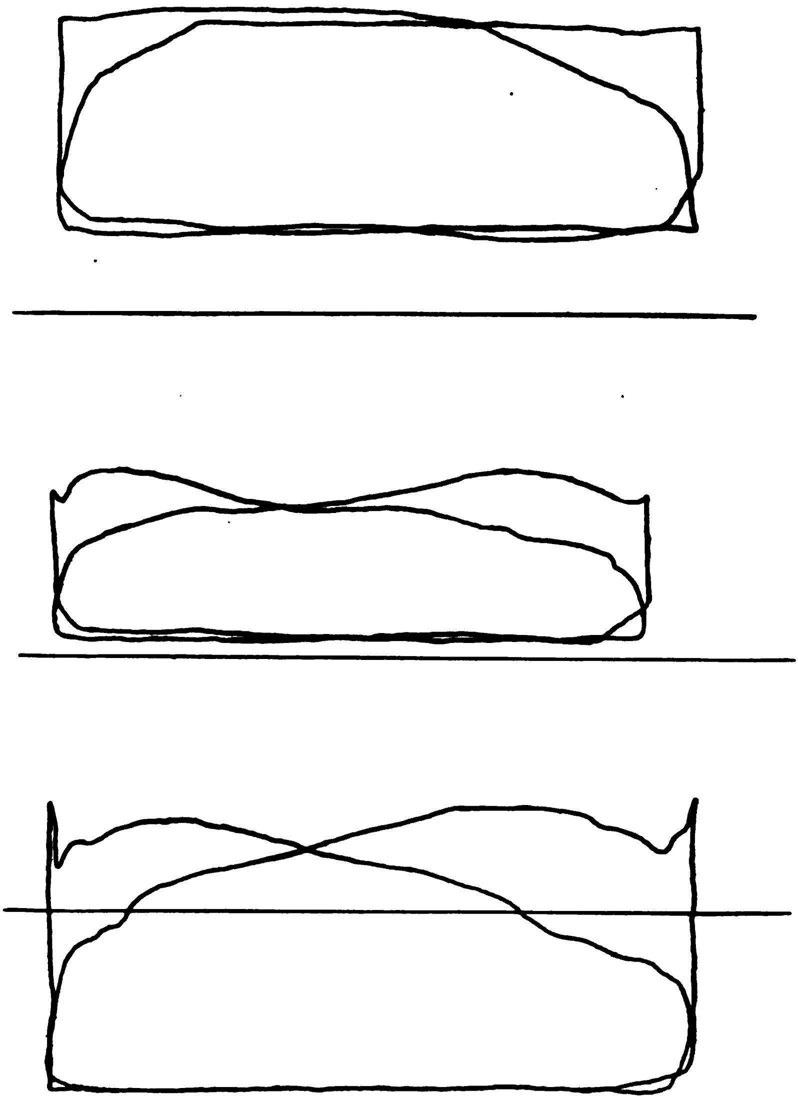




M. E. P.	I. H. P.		
H = 123.6	H = 2,602.	Steam	220.
M = 41.7	M = 2,525.	M. P. Rec.	59.
L = 12.81	L = 2,605.	L. P. Rec.	10.5
	Total, 7,732.	Vacuum	25.5″
		R. P. M.	75.5
		Piston Speed	79.3

Cut off: H. P.=.71, M. P. & L. P.=Full

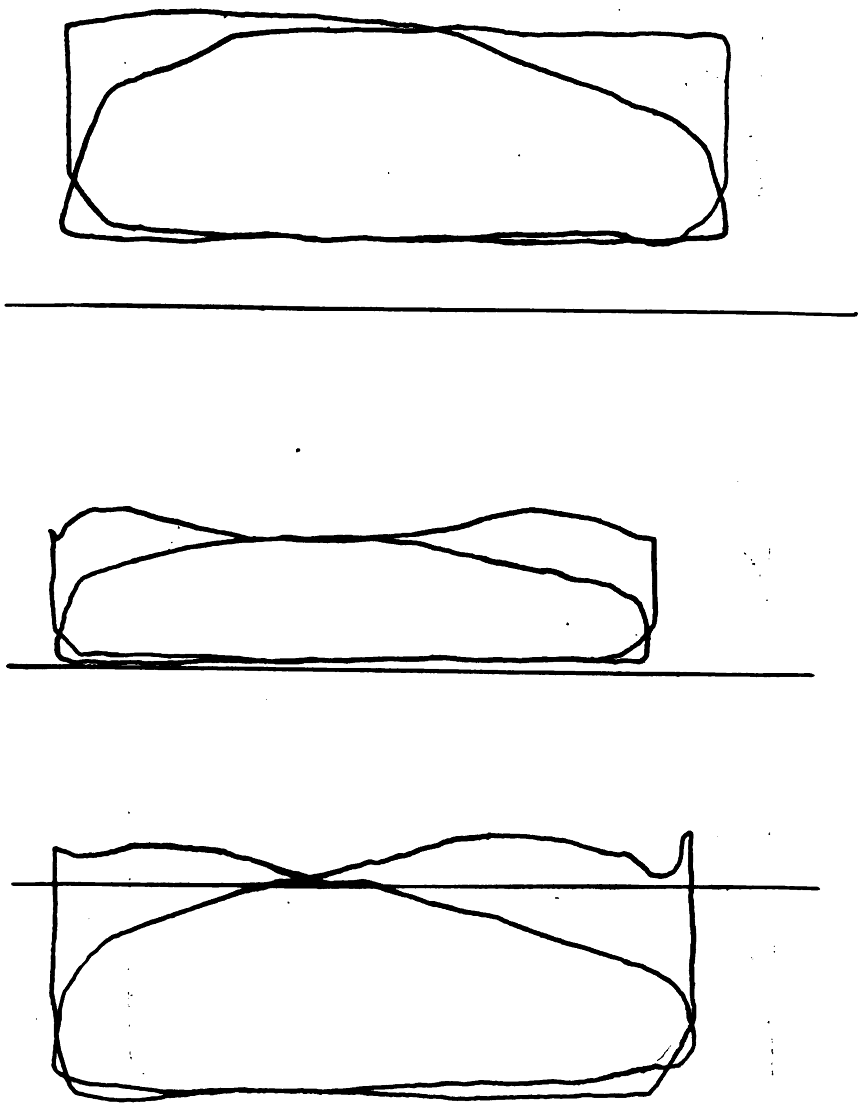
Series 4



M. E. P.	I. H. P.		
H = 122.55	H = 2,561.84	Steam	220.
M = 41.10	M = 2,471.60	M. P. Rec.	61.
L = 11.74	L = 2,360.95	L. P. Rec.	10.5
	Total, 7,394.39	Vacuum	25.″
		R. P. M.	75.
		Piston Speed	787.5

Cut off: H. P. = .69, M. P. & L. P. = .75

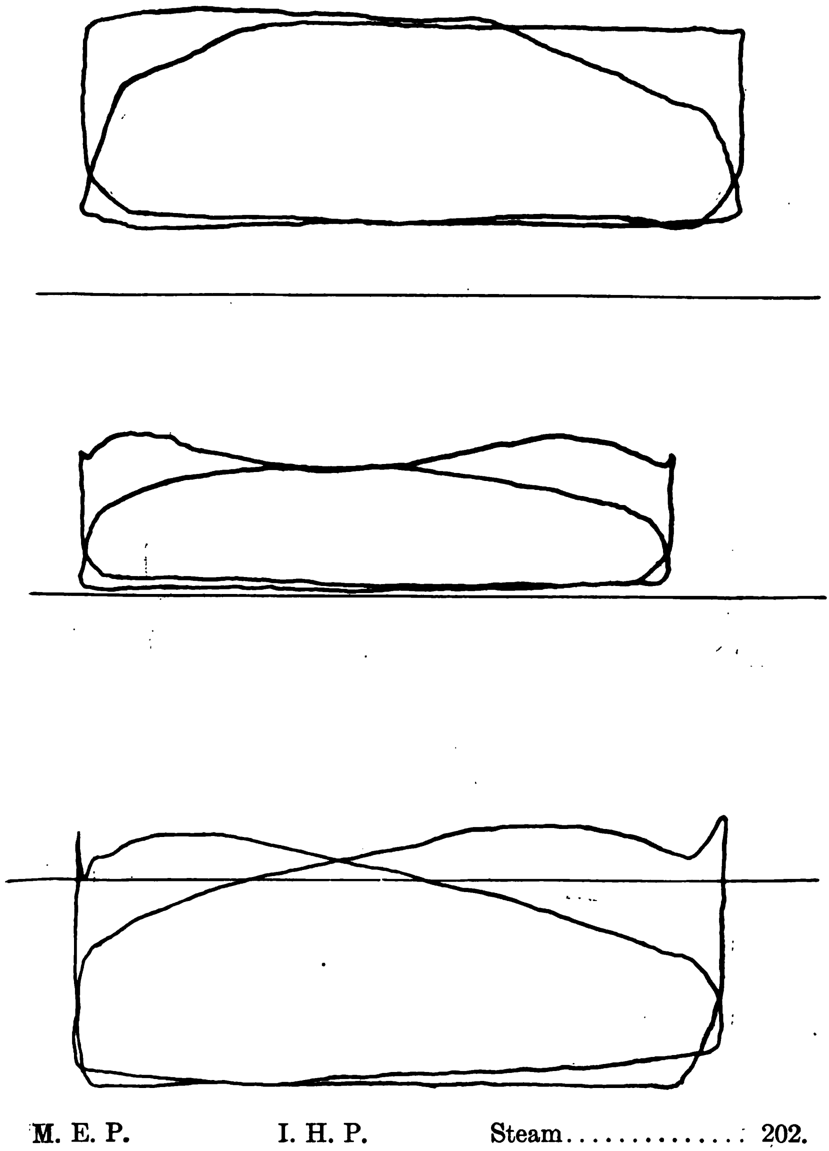
SERIES 4



M. E. P.	I. H. P.		
H = 125.1	H = 2,553.7	Steam..............	210.
M = 36.9	M = 2,167.5	M. P. Rec.	49.5
L = 10.48	L = 2,067.2	L. P. Rec..........	8.0
	Total, 6,788.4	Vacuum	25.″
		R. P. M............	73.25
		Piston Speed	769.125

Cut off: H. P.=.66, M. P.—.75, L. P.=.735

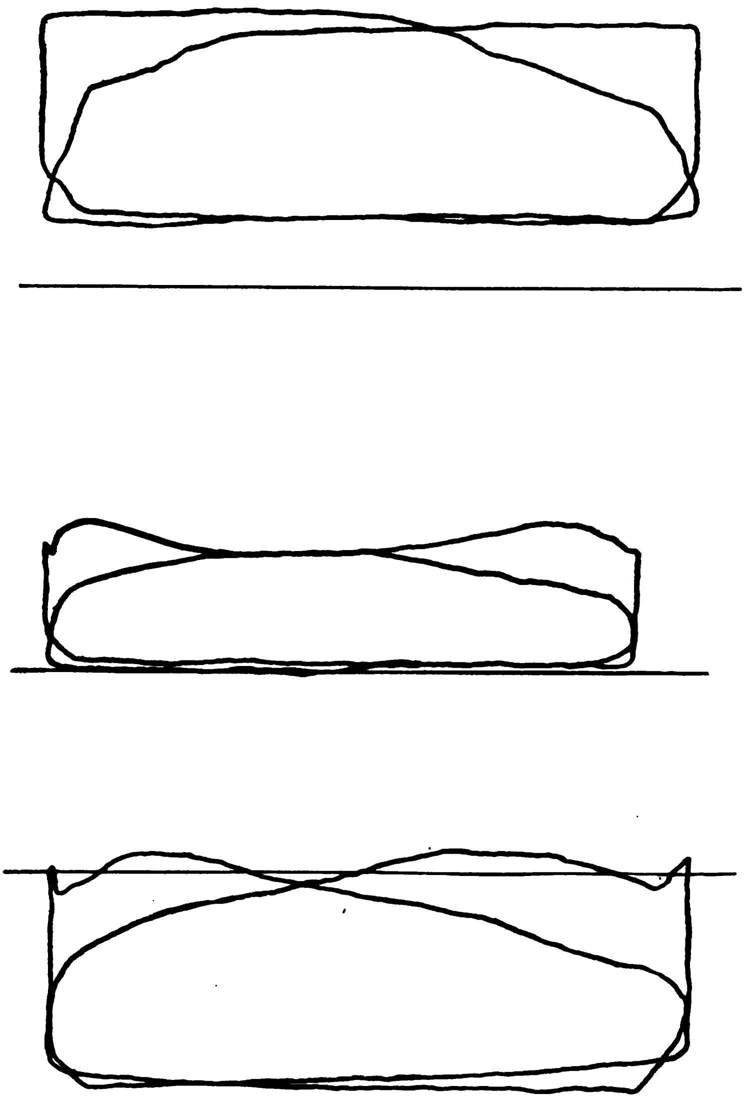
SERIES 4



M. E. P.	I. H. P.		
H = 114.3	H = 2,301.66	Steam..............	202.
M = 36.0	M = 2,085.48	M. P. Rec.	50.
L = 10.77	L = 2,095.41	L. P. Rec.	8.
	Total, 6,482.55	Vacuum	25.″
		R. P. M..............	72.25
		Piston Speed	758.6

Cut off: H. P.=.69, M. P.=.73, L. P.=.75

Series 4



M. E. P.	I. H. P.		
H = 110.55	H = 2,141.3	Steam	204.
M = 34.42	M = 1,918.3	M. P. Rec.	45.
L = 9.75	L = 1,828.4	L. P. Rec.	6.5
	Total, 5,888.0	Vacuum	25.5″
		R. P. M.	69.5
		Piston Speed	729.75

Cut off: H. P. = Normal, Throttled, M. P. & L. P. = Full

TRIAL TRIP OF PASSENGER STEAMER AT DELAWARE BREAKWATER

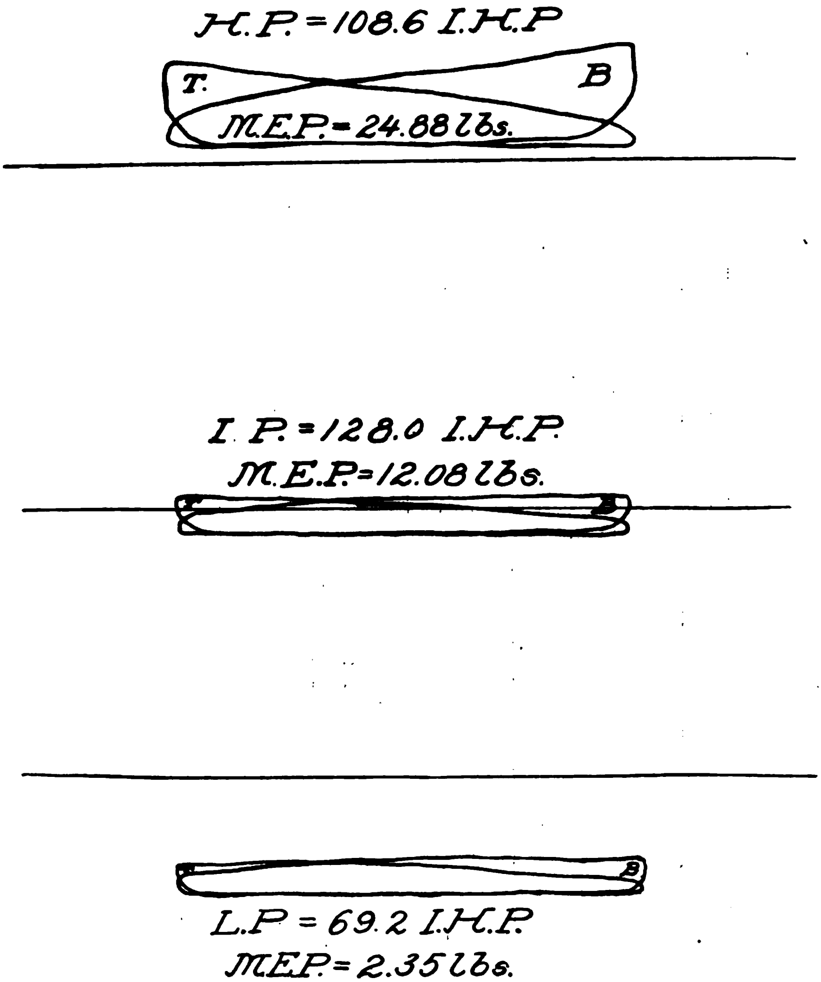
NO. OF RUN	STARBOARD ENGINE						PORT ENGINE						TOTAL I. H. P.
	STEAM	1ST REC.	2D REC.	VAC. INS.	REV.	I. H. P.	STEAM	1ST REC.	2D REC.	VAC. INS.	REV.	I. H. P.	
1	145	18	-9	26	99	305.8	145	17	-7½	26½	96	308.4	614.2
2	144	17	-8½	25½	96½	392.5	145	16½	-7½	26½	97½	305.2	697.7
3	133	33½	0	25	118	739.2	133	35	1	26	119	656.5	1,395.7
4	137	34½	0	26	122	711.8	137	33½	0	27	123	740.1	1,451.9
5	147	51	5½	25½	136	No Cards	147	54	3	26½	136	No cards	Taken
6	150	52	6	26	136	No Cards	150	54	3	26	137	No cards	Taken
7	155	55	7	26	142	1,126.1	155	56	5	26	143	1,111.5	2,237.6
8	158	57	10	26	146½	1,192.3	158	57	7	26	146½	1,216.9	2,409.2

Scale of Springs used: H. P. = 80 lbs. M. P. = 30 lbs. L. P. = 16 lbs.

Length of course = 1.261 nautical miles.

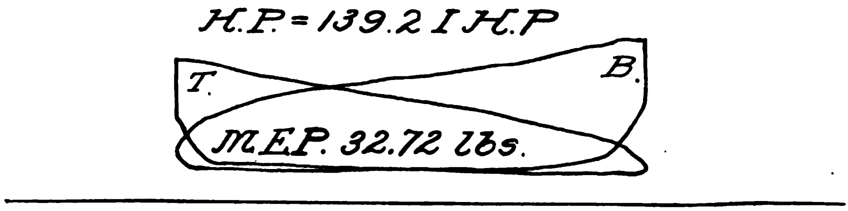
Engine $\frac{19\frac{1}{4}'' \times 30'' \times 50''}{30''}$

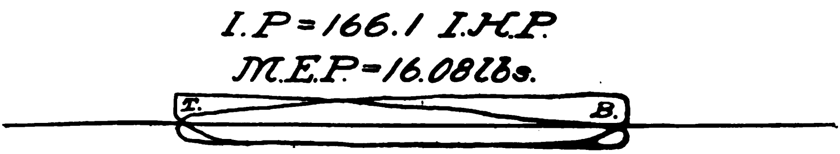
SERIES 5

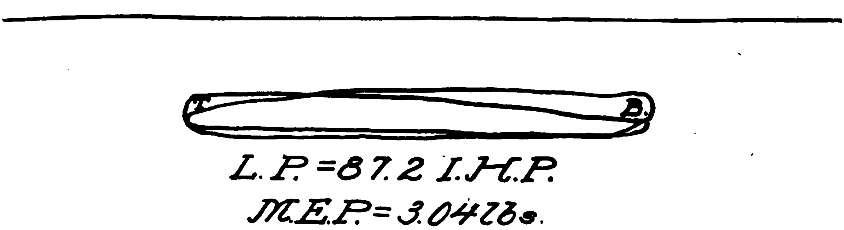


No. 1 STARBOARD

SERIES 5

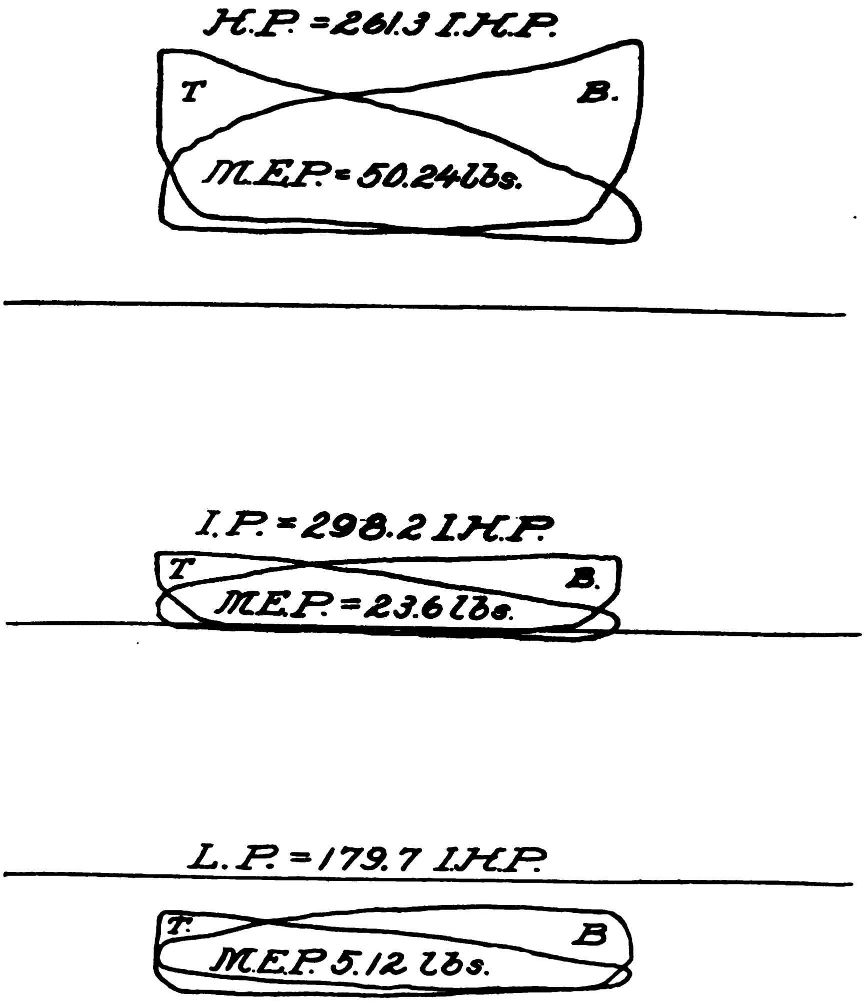






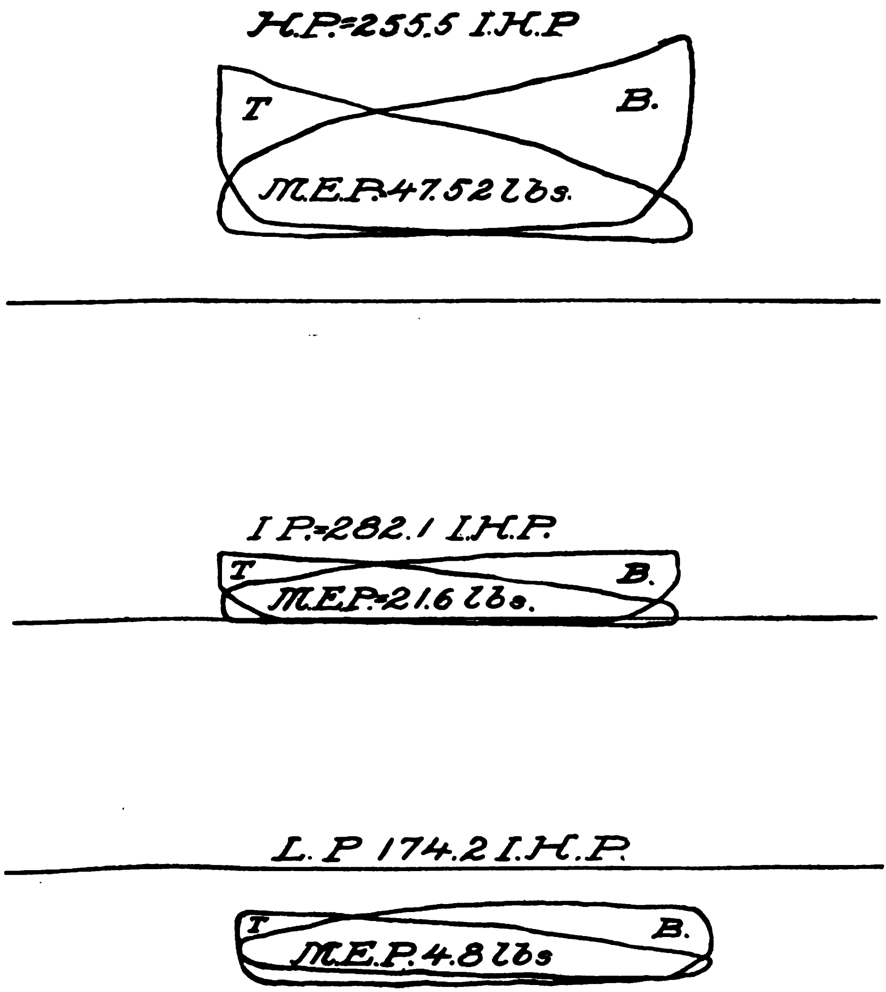
No. 2 STARBOARD

Series 5



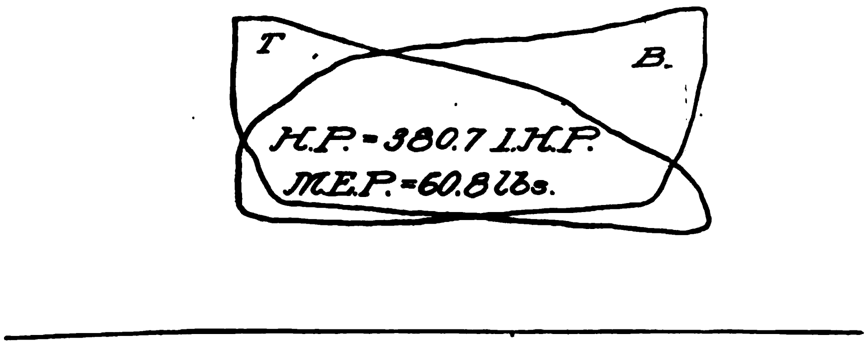
No. 3 Starboard

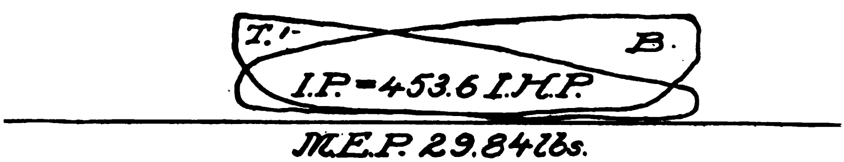
SERIES 5

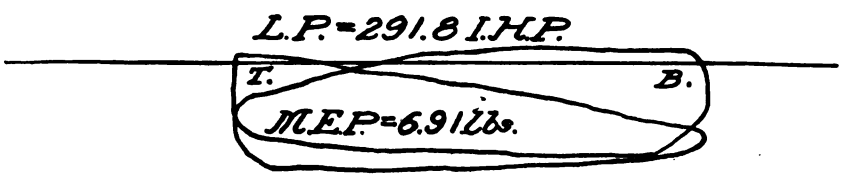


No. 4 STARBOARD

SERIES 5

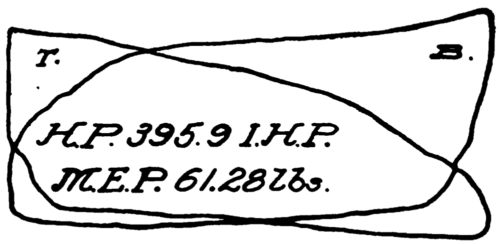


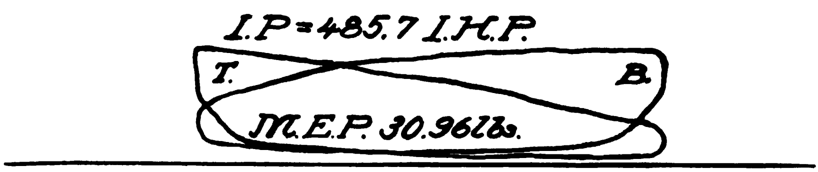


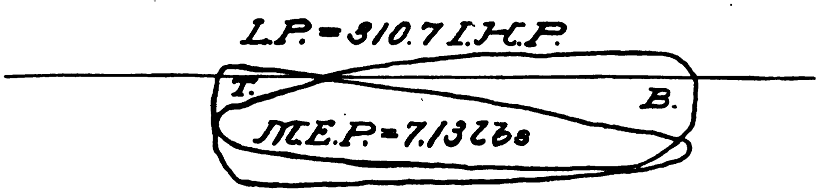


NO. 7 STARBOARD

SERIES 5

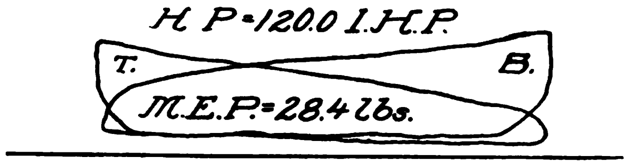


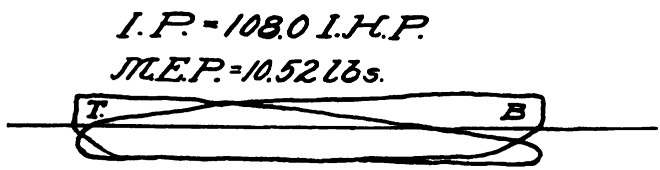


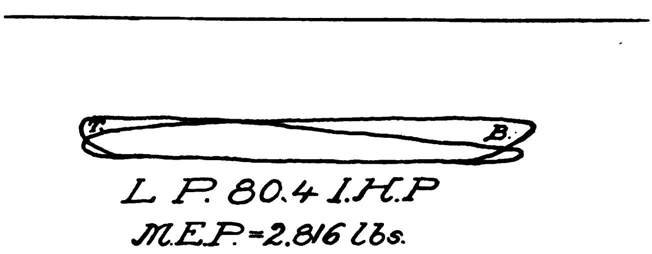


No. 8 STARBOARD

Series 5

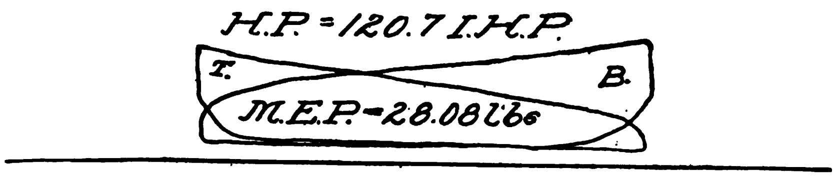


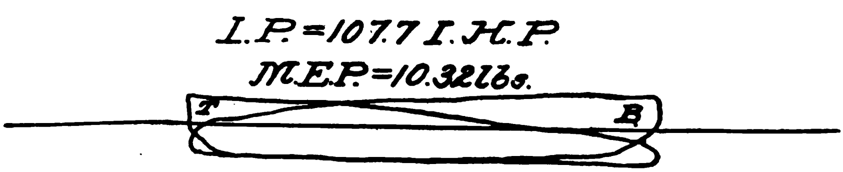


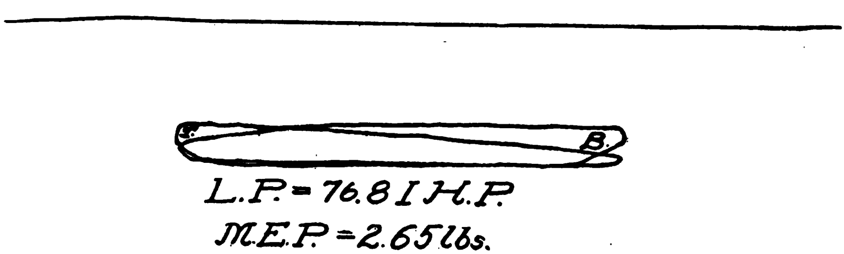


No. 1 Port

SERIES 5

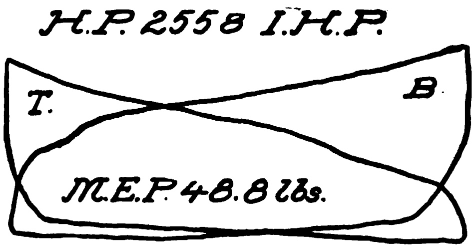


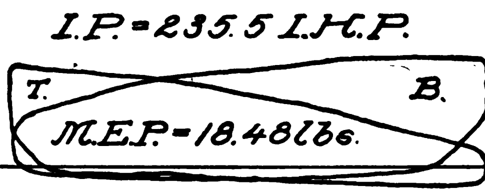


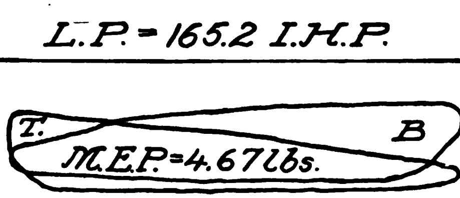


No. 2 PORT

SERIES 5

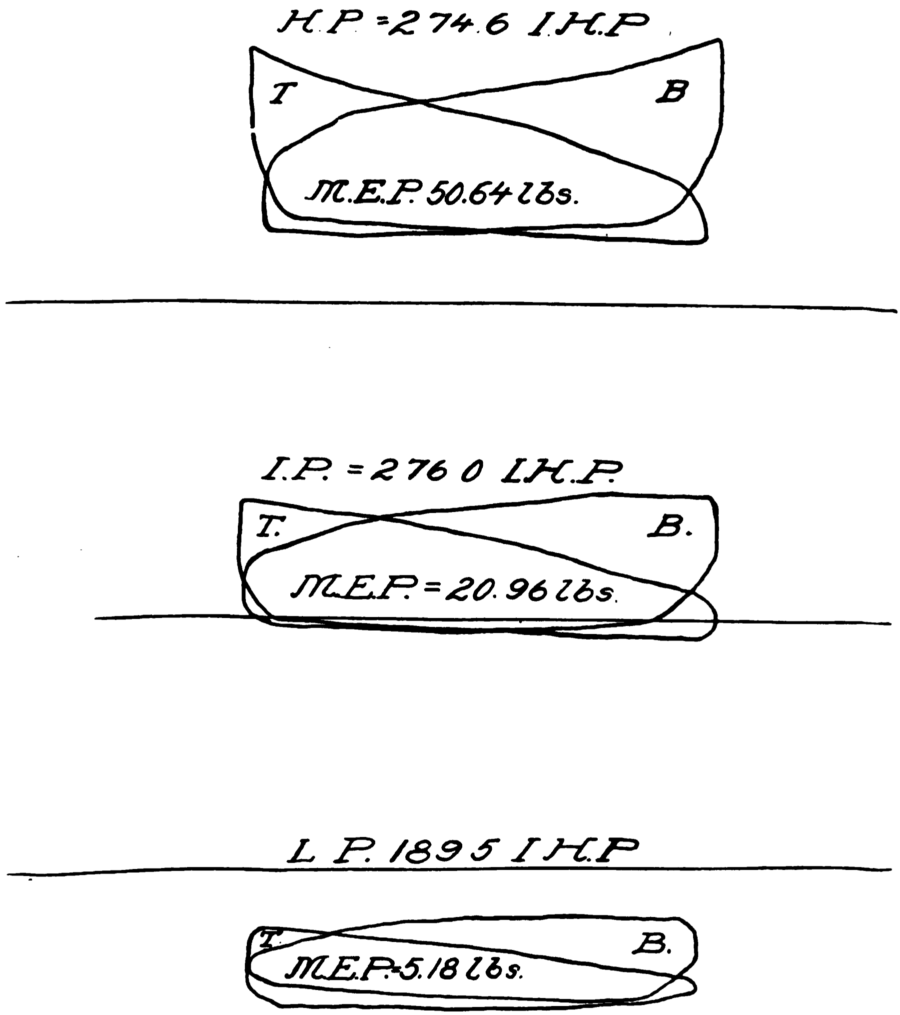






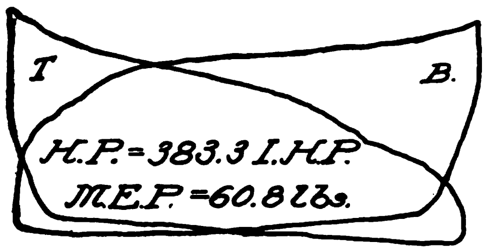
No. 3 PORT

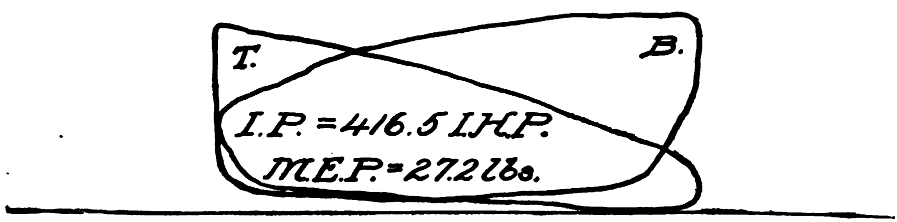
SERIES 5

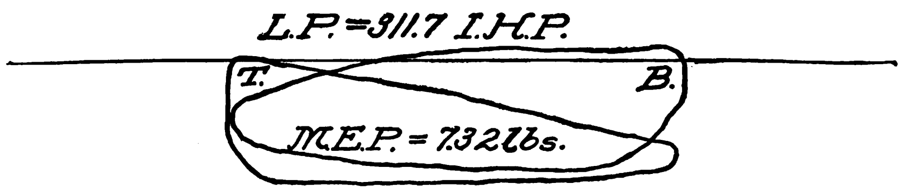


No. 4 PORT

Series 5

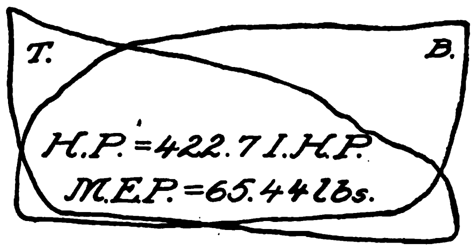


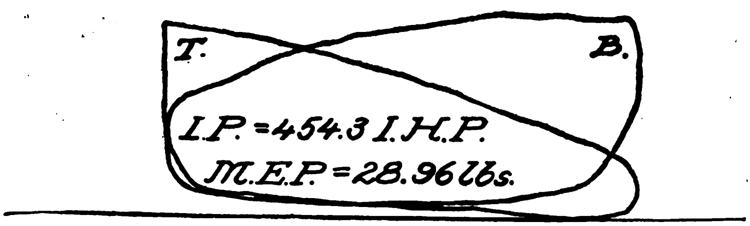


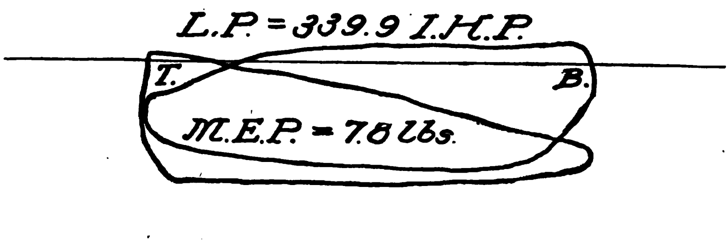


No. 7 Port

Series 5



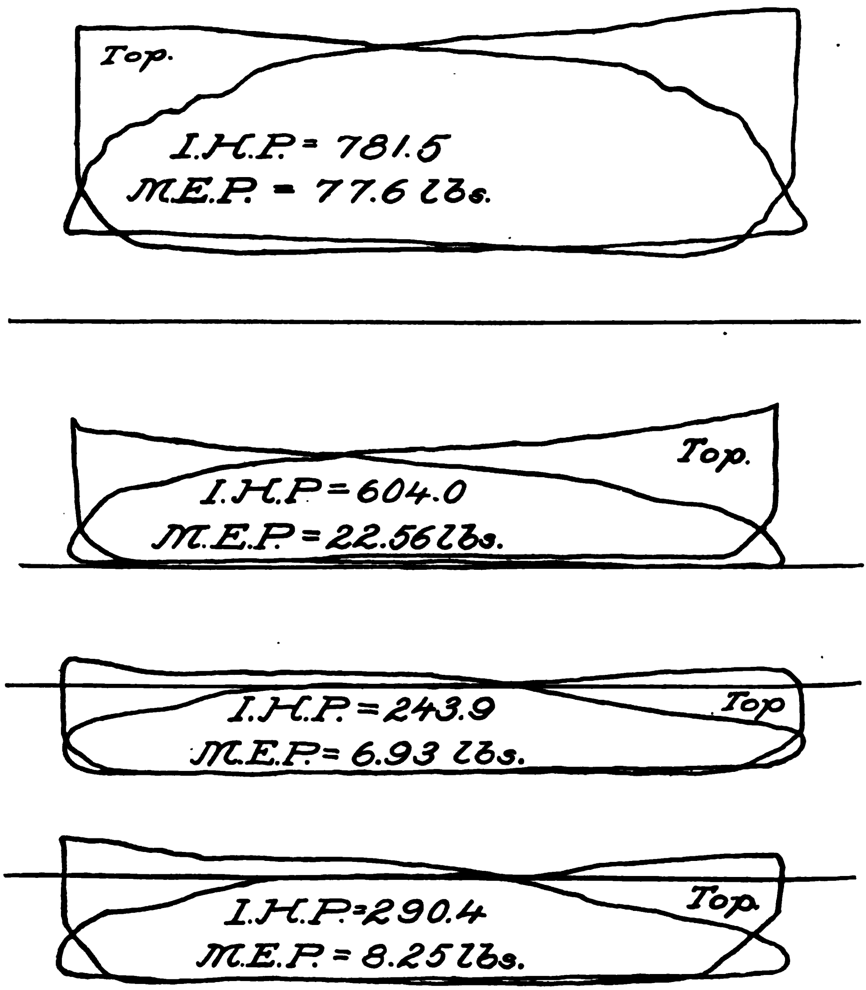




No. 8 Port

Series 6

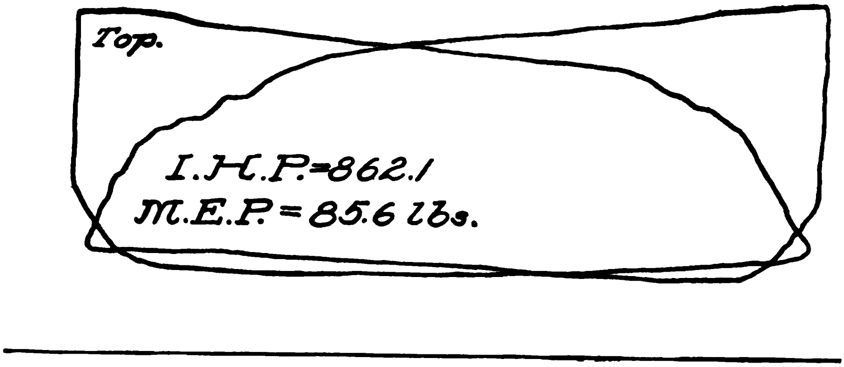
INDICATOR DIAGRAMS TAKEN FROM ENGINE $\frac{23'' \times 37\frac{1}{2}'' \times 43'' \times 43''}{30''}$

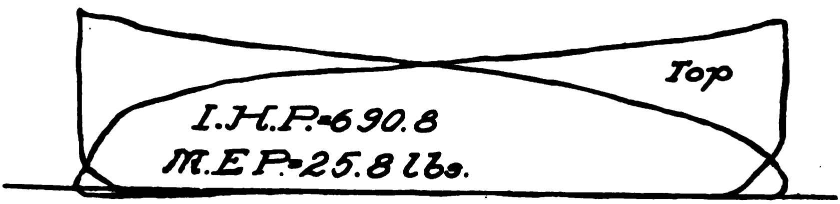


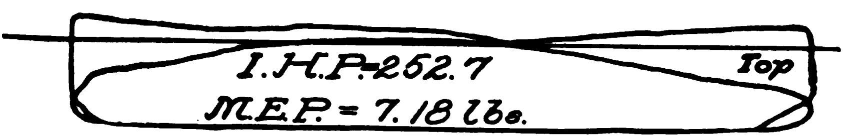
No. 1 Starboard

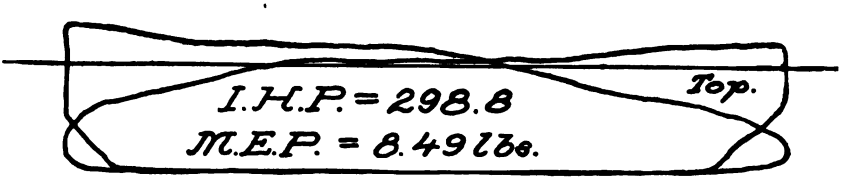
Steam 145 lbs. 1st Rec. 38 lbs. 2d Rec. 3 lbs. Vac. 22″ Rev. 160
I. H. P., H. P. 781.5 I. H. P., I. P. 604.0
I. H. P., F. L. P. 243.9 I. H. P., A. L. P. 290.4 Total I. H. P. 1919.8
Scale of springs used: H. P. = 80 lbs., M. P. = 30 lbs., L. P. = 16 lbs.

SERIES 6







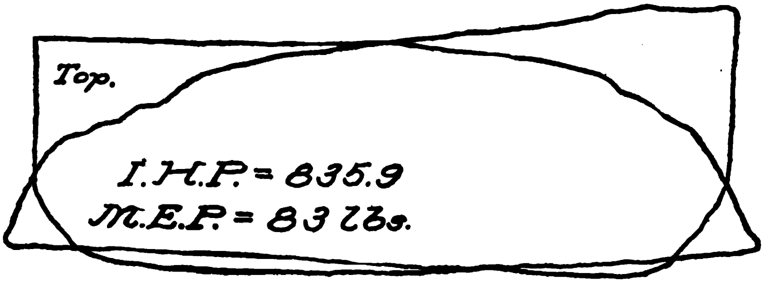


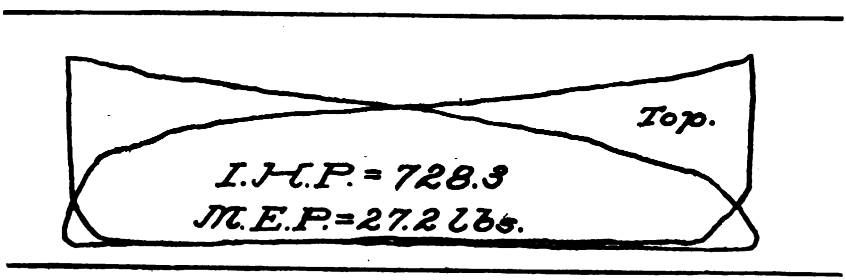
No. 2 STAR. ENG.

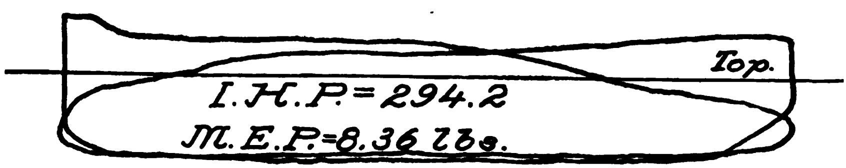
Rev. 160

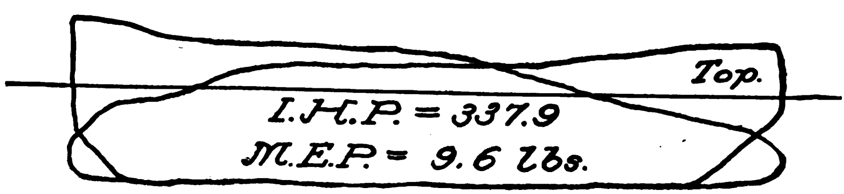
I. H. P., H. P. 862.1	I. H. P., I. P. 690.8	
I. H. P., F. L. P. 252.7	I. H. P., A. L. P. 298.8	Total, 2,104.4

SERIES 6







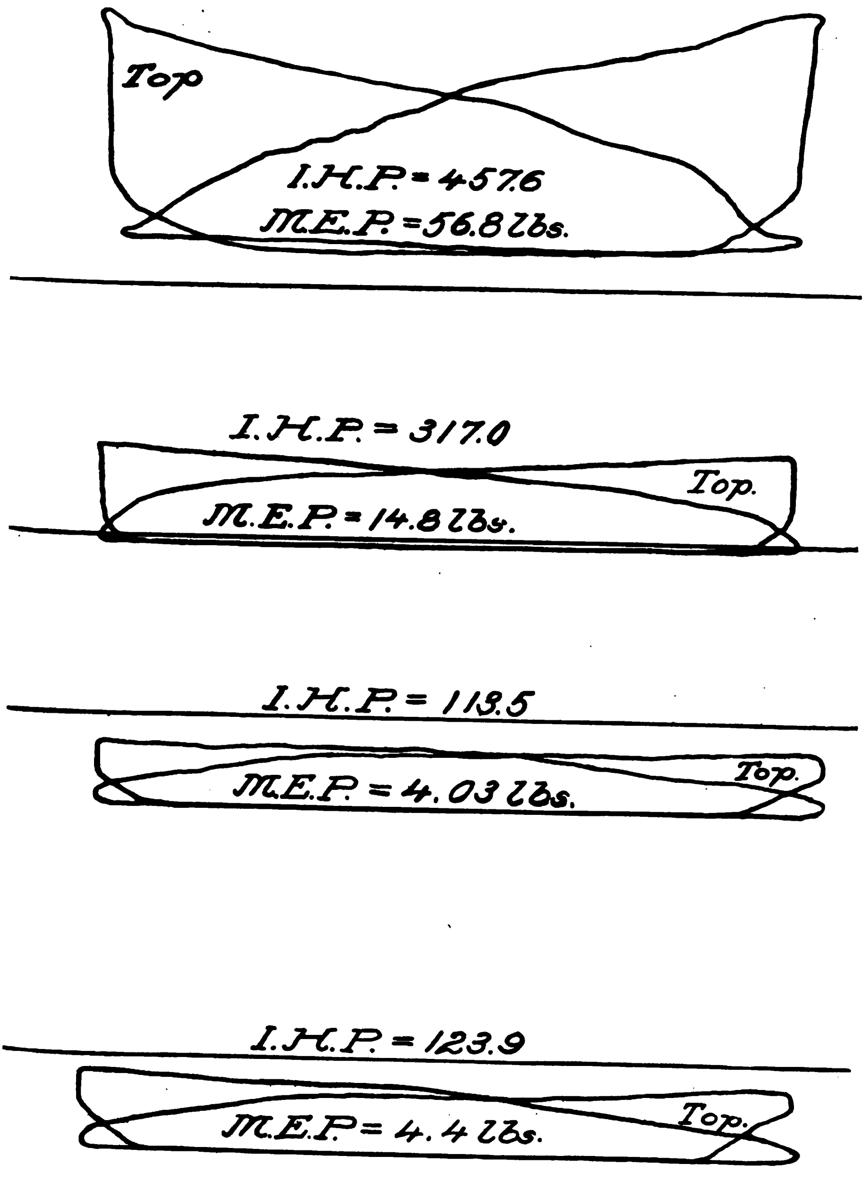


NO. 3 STAR. ENG.

Rev. 160

I. H. P., H. P. 835.9	I. H. P., I. P. 728.3	
I. H. P., F. L. P. 294.2	I. H. P., A. L. P. 337.9	Total, 2,196.3

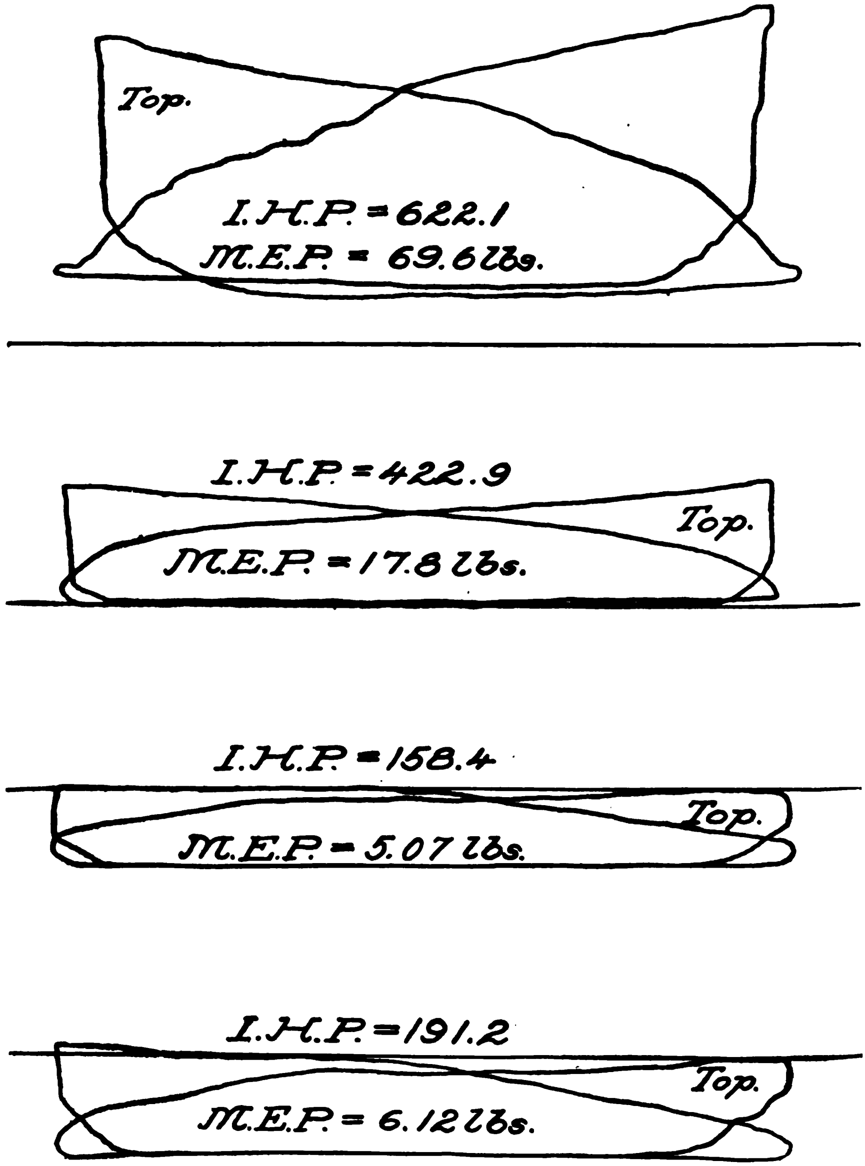
SERIES 6



No. 4 STAR ENG.

Steam 150 1st Rec. 20 2d Rec. –5 Vac. 21″ Rev. 128
I. H. P., H. P. 457.6 I. H. P., I. P. 317.0
I. H. P., F. L. P. 113.5 I. H. P., A. L. P. 123.9 Total, 1,012.0

SERIES 6



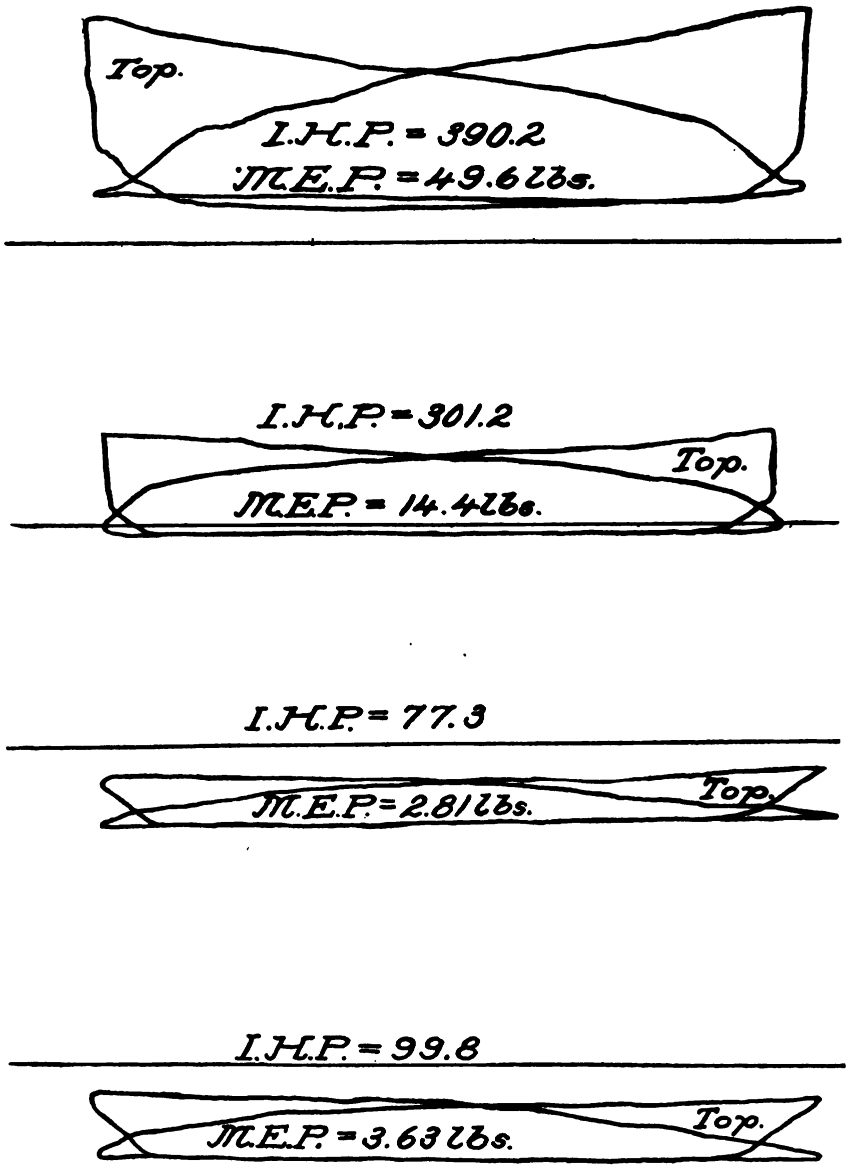
No. 5 STAR. ENG.

Steam 149 1st Rec. 30 2d Rec. 2 Vac. 21″ Rev. 142

I. H. P., H. P. 622.1 I. H. P., I. P. 422.9

I. H. P., F. L. P. 158.4 I. H. P., A. L. P. 191.2 Total, 1,394.6

SERIES 6



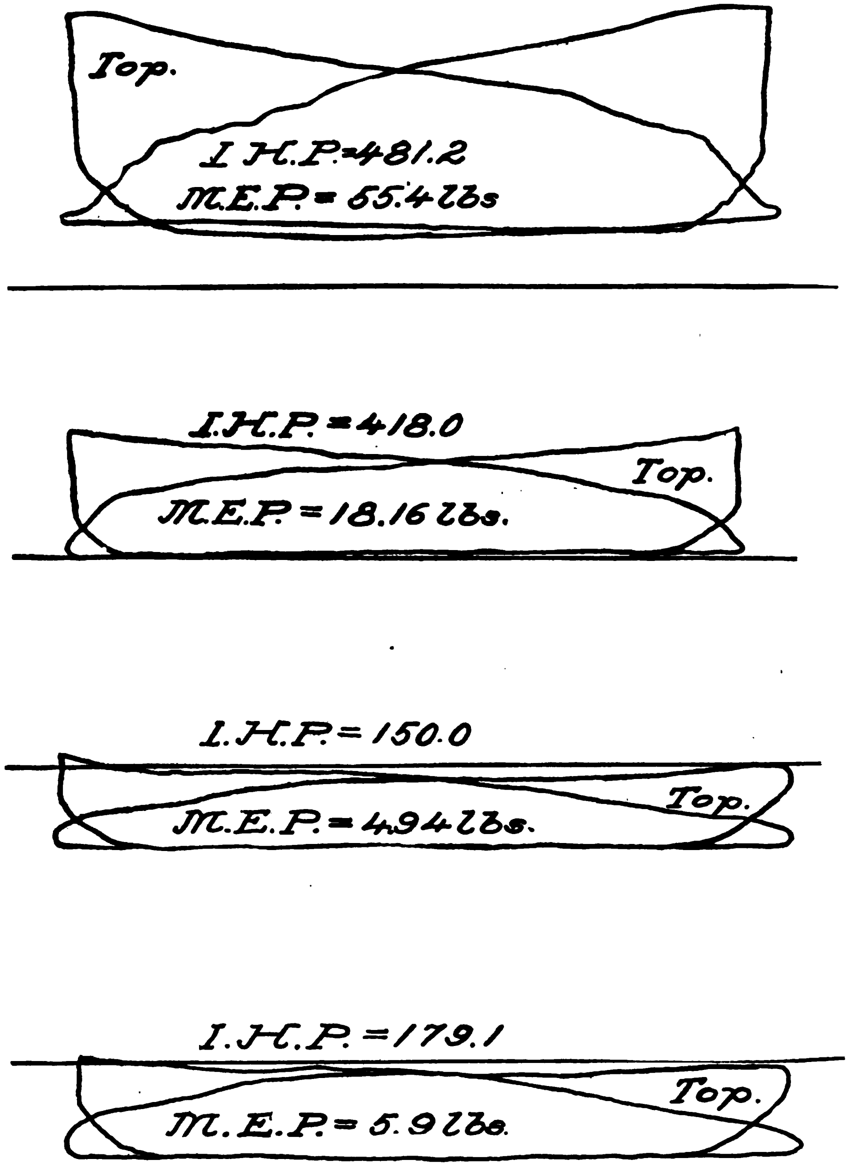
No. 6 PORT ENG.

Steam 125 1st Rec. 20 2d Rec. 6 Vac. 21″ Rev. 125

I. H. P., H. P. 390.2 I. H. P., I. P. 301.2

I. H. P., F. L. P. 77.3 I. H. P., A. L. P. 99.8 Total, 868.5

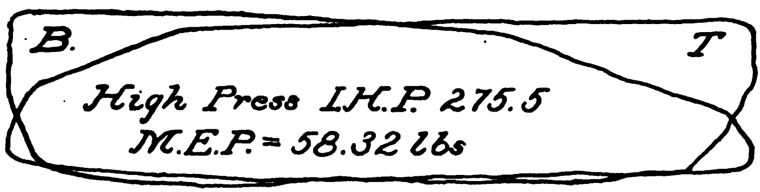
SERIES 6



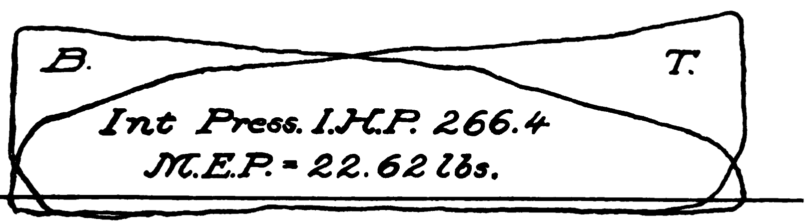
No. 7 PORT ENGINE

Steam 143 1st Rec. 30 2d Rec. 2 Vac. 21″ Rev. 138
I. H. P., H. P. 481.2 I. H. P., I. P. 418.0
I. H. P., F. L. P. 150.0 I. H. P., A. L. P. 179.1 Total, 1228.3

SERIES 7

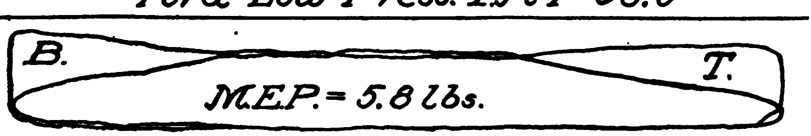


80 lbs. Spring.



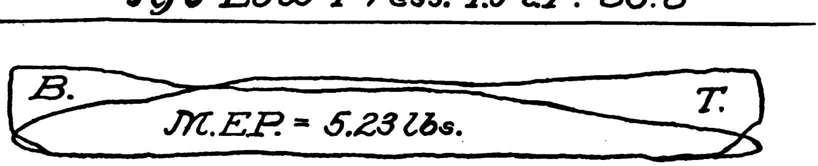
30 lbs. Spring.

Ford Low Press. I.H.P 93.0



16 lbs. Spring.

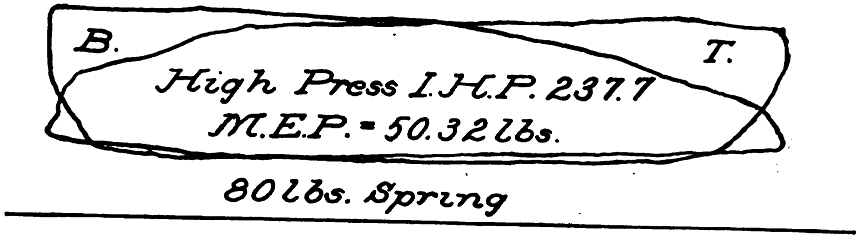
Aft Low Press. I.H.P. 83.8

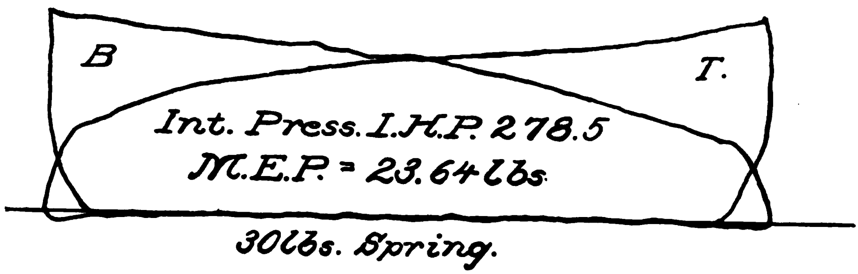


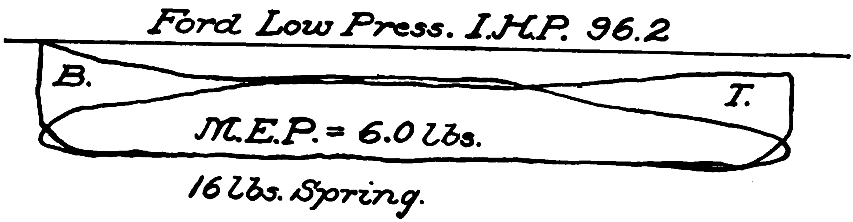
16 lbs. Spring.

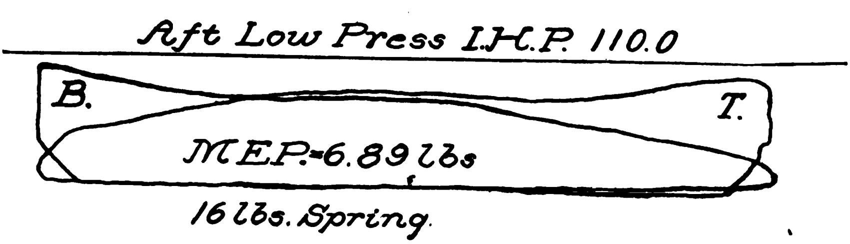
No. of Run	Steam	1st Rec.	2d Rec.	Vac.	Rev.
1	115	33	−1	$24\frac{1}{2}''$	110

SERIES 7



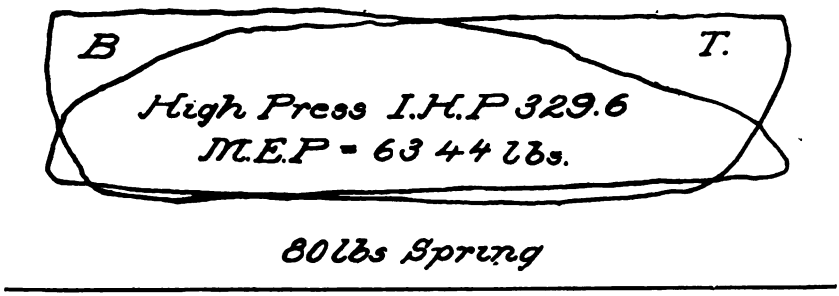


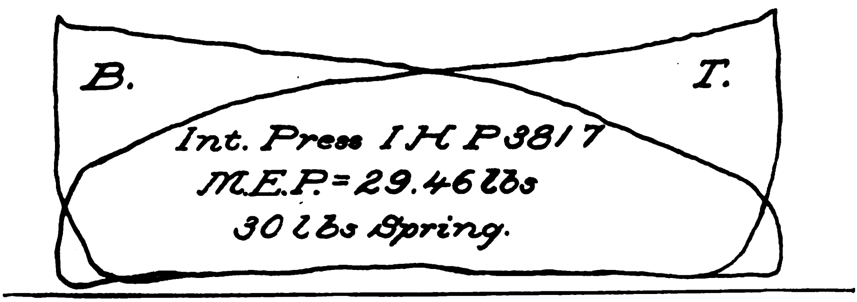


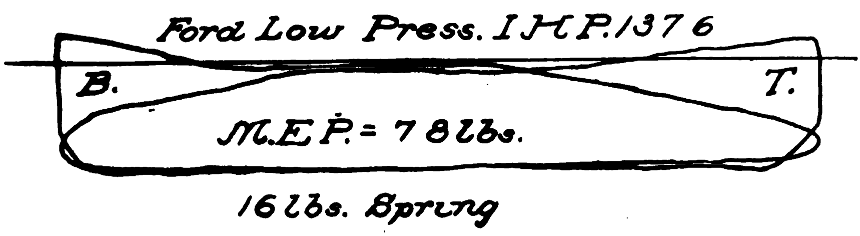


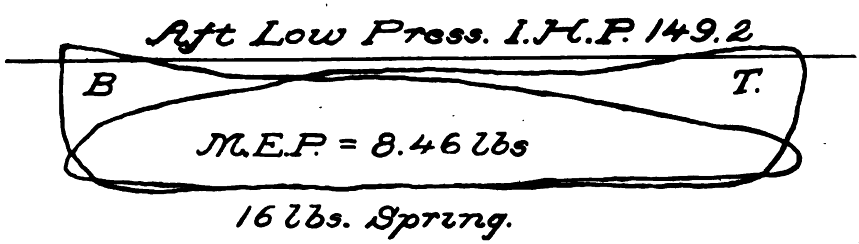
No. of Run	Steam	1st Rec.	2d Rec.	Vac.	Rev.
2	107½	34	−1	24½″	110

SERIES 7



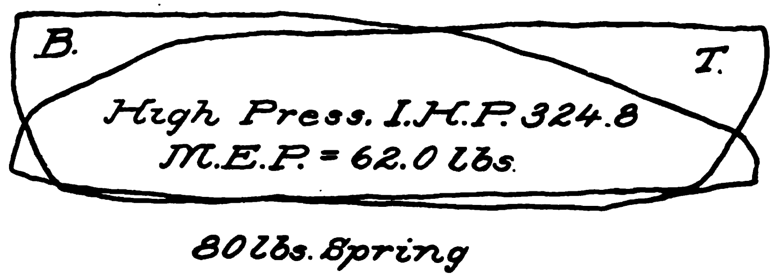


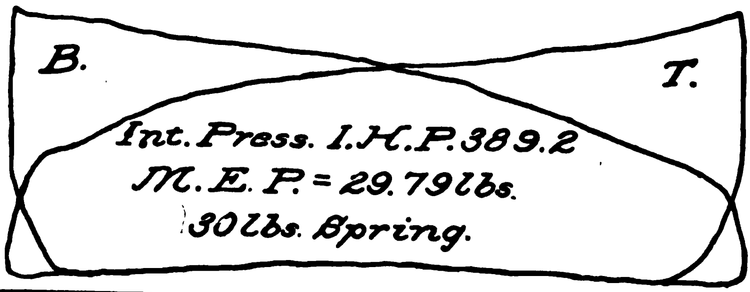


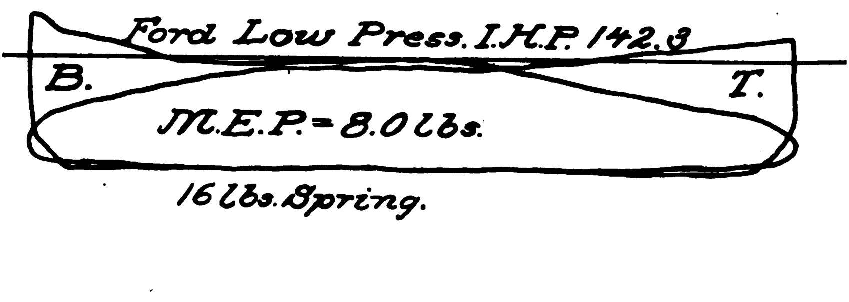


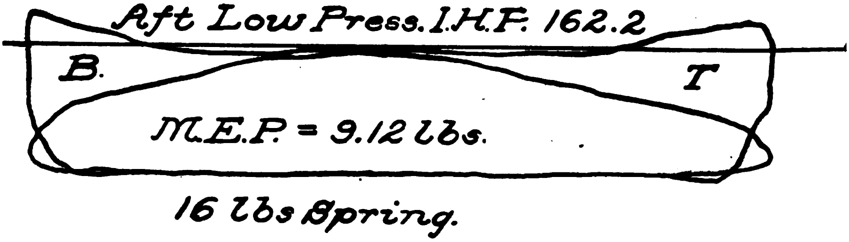
No. of Run	Steam	1st Rec.	2d Rec.	Vac.	Rev.
3	137½	45	3½	24½″	121

SERIES 7



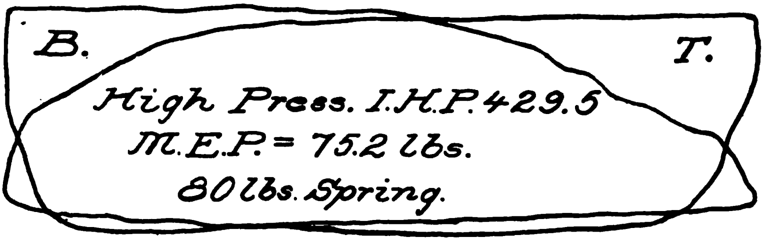


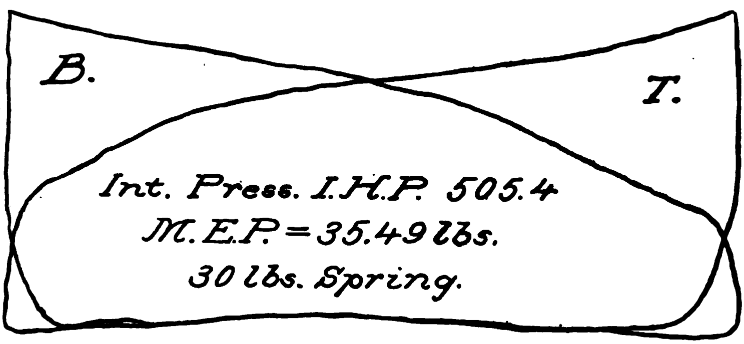


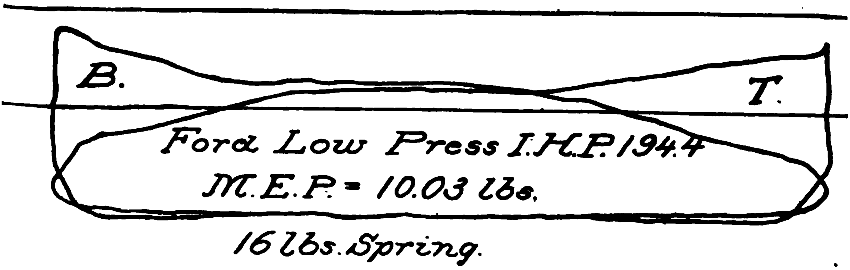


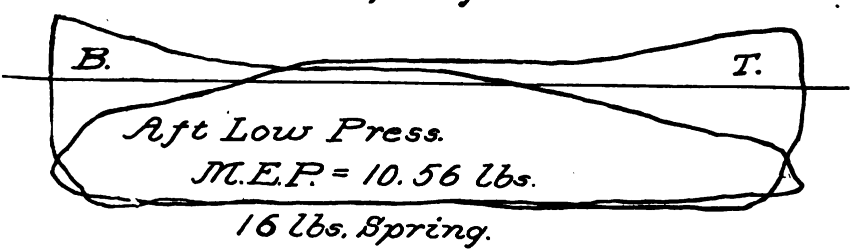
No. of Run	Steam	1st Rec.	2d Rec.	Vac.	Rev.
4	135	46½	3¾	24½″	122

SERIES 7



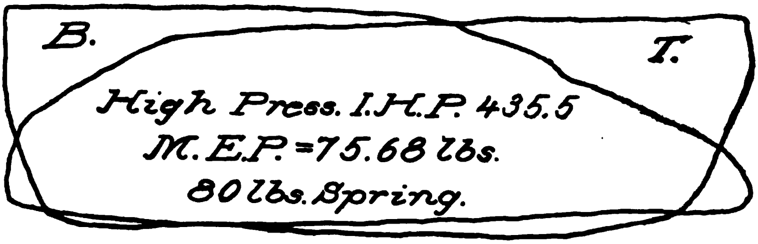


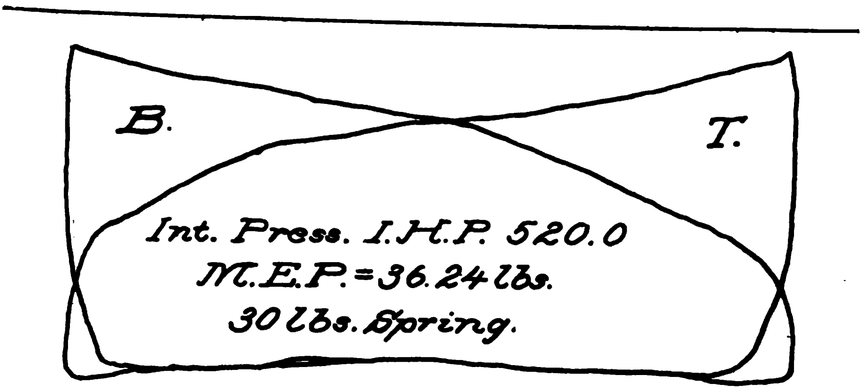


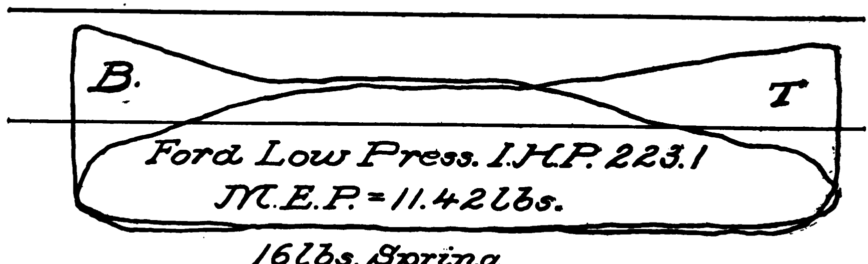


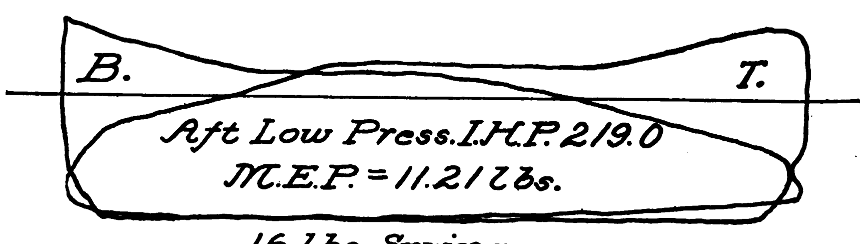
No. of Run	Steam	1st Rec.	2d Rec.	Vac.	Rev.
5	168	57	8	24″	133

SERIES 7









No. of Run	Steam	1st Rec.	2d Rec.	Vac.	Rev.
6	170	59	10	25″	134

LOG OF TRIAL TRIP OF JANUARY 24, 1907

FREIGHT STEAMER "TUSCAN"	POINTS DOWN	TIME	Elapsed Time Minutes	Nautical Miles	Statute Miles	Speed Knots	Speed Miles	Mean Rev's	Mean Steam	Mean I H.P	Slip %
Place, Patapsco River and Chesapeake Bay	Passed Sandy Point	11-12-13									
270 Tons Coal			32′, 2″	7.4	8.5	13.8	15.87	81.6	182.6	2838	21
	" Thomas Point	11-44-15									
			14′, 14″	3.25	3.74	13.68	15.73				
F. W. Tanks Full	" Bloody Point	11-58-29									
Draft on Trial { For'd.... 9′, 9″ Aft......13′, 6″ Mean ...11′, 7½″	" Bloody Point	12-07-00									
			11′, 45″	3.25	3.74	16.6	19.0				
	" Thomas Point	12-18-45									
Wetted Surface, 12,800 Square Feet			27′, 15″	7.4	8.5	16.26	18.69	86.4	185.3	3079	13
	" Sandy Point	12-46-00									
I.H.P. per 100 square inches, W. S. @ 10 knots, 6.6	" Sandy Point	12-55-00									
			30′, 03″	7.4	8.5	14.76	16.97	83.5	176.5	3059	19
Displacement, 2,260 Tons	" Thomas Point	1-25-03									
	" Thomas Point	1-35-00									
Admirality Co-eff., 200.9			28′, 15″	7.4	8.5	15.72	18.07	84.6	170.3	2983	14
	" Sandy Point	2-03-15									
Fore Peak Tank Full to 1 ft. of Lower Deck	" Sandy Point	2-10-00									
Aft. Tank Full			30′, 15″	7.4	8.5	14.64	16.83	85.5	184	3252	20
	" Thomas Point	2-40-15									
	Thomas Point	2-59-30									
			27′, 30″	7.4	8.5	16.14	18.56	83.3	180.6	2928	15
	Sandy Point	3-27-00									

	No. Card	Time	Steam	Vac.	M. P. Rec.	L. P. Rec.	Rev.	M.R.P.	I. H. P. H. P.	I. H. P. M. P.	I. H. P. L. P.	I. H. P. Total
C	1	10–45	180	27	66	14	83.3	33.0	823.27	870.21	1090.91	2784.39
C	2	11–00	186	27	67	16	83.3	34.0	840.66	805.72	1216.79	2863.17
O	3	11–15	184	27	70	19	85	34.2	780.91	855.05	1305.84	2941.80
C	4	11–30	185	27	67	16	82	33.4	821.84	793.15	1156.50	2771.49
C	5	11–45	183	27½	65	14	84.5	32.8	846.89	749.11	1106.63	2802.63
O	6	12–00	188	27	72	19	85	35.5	798.66	904.38	1348.66	3051.70
O	7	12–15	183	27	72	20	83	34.4	693.22	867.04	1421.44	2981.70
O	8	12–30	188	27	80	22½	85.6	35.2	643.50	894.20	1509.08	3046.78
O	9	12–45	185	27	78	21½	85	37.4	615.26	1118.14	1477.10	3210.50
O	10	1–00	178	27	74	20	82.5	37.0	597.17	1089.50	1412.89	3099.56
O	11	1–15	175	27	72	20	83.3	35.8	602.96	1031.32	1384.62	3018.90
O	12	2–15	178	27	77	23	85	37.4	427.11	1141.03	1541.32	3209.46
O	13	2–30	190	27	77	21	86	37.9	682.36	1098.02	1516.13	3296.51
C	14	2–45	182	27	66	16	82.6	33.6	773.86	894.82	1141.15	2809.83
C	15	3–00	176	27½	63	14½	82.6	31.9	735.87	846.89	1081.76	2664.52
O	16	3–15	183	27	72	20½	84	35.6	689.88	942.49	1396.26	3028.73
O	17	3–30	183	27	73	21	83.5	36.6	627.67	991.49	1472.06	3091.22

Card marked "C" by pass closed. Cards marked "O" by pass open.

1st Mean					2nd Mean					3rd Mean					4th Mean				
Steam	Rev.	I.H.P.	Speed Knots	Speed Miles	Steam	Rev.	I.H.P.	Speed Knots	Speed Miles	Steam	Rev.	I.H.P.	Speed Knots	Speed Miles	Steam	Rev.	I.H.P.	Speed Knots	Speed Miles
182.6	81.6	2838	13.8	15.87	183.9	84	2958	15.03	17.28										
185.3	86.4	3079	16.26	18.69						178.6	84	2989	15.13	17.40					
176.5	83.5	3059	14.76	16.97	173.4	84	3021	15.24	17.52						178.2	84.1	3022	15.22	17.51
170.3	84.6	2983	15.72	18.07						177.8	84.2	3055	15.31	17.63					
184	85.5	3252	14.64	16.83	182.3	84.4	3090	15.39	17.74										
180.6	83.3	2928	16.14	18.56															

LOG OF TRIP OF JANUARY 28 AND 29, 1907—BALTIMORE TO PHILADELPHIA

Freight Steamer "Tuscan"	Points Down	Time	Elapsed Time Minutes	Nautical Miles	Statute Miles	Speed Knots	Speed Miles	Mean Rev's	Mean Steam	Mean I.H.P.	Slip %
Place, Patapsco River and Chesapeake Bay	Passed Sandy Point	3-14-45									
310 Tons Coal	" Thomas Point	3-45-30	30', 43"	7.4	8.5	14.44	16.60	81.6	170	2480	18
F. W. Tanks Full	" Sharps Island	4-52-20	66', 50"	15.3	17.6	13.72	15.79	79.2	171	2472	20
Draft { For'd.....10', 4"	" Coye Point	5-52-50	60', 30"	15.2	17.5	14.76	16.97	80.4	174		14
Aft.......13', 8" Mean ....12', 0"	" Cedar Point	6-13-30	20', 40"	5.2	6.0	15.05	17.30	81.6	178.3		15
Wetted Surface, 13,000 Square Inches	" Point Lookout	7-17-00	63', 30"	15.4	17.7	14.54	16.72	81.	180		17
I.H.P. per 100 square inches, W. S., @ 10 knots, 6.6	" Smith's Point.......	8-00-50	43', 50"	10.7	12.3	14.62	16.81	77.4	170.6		15
Displacement, 2,340 Tons	" Wind Mill	9-11-00	70', 10"	17.0	19.5	14.51	16.69	78.6	165		15
Admiralty Co-eff., 202.2	" Wolf Trap	10- 5-45	54', 45"	12.2	14.0	13.36	15.36	79.2	175		22
Fore Peak Tank Full to 1 ft. of Lower Deck	" York Spit	10-53-20	47', 35"	10.6	12.2	13.21	15.19	80.4	179		23
Aft. Tank Full											

READINGS OF TRIP, JANUARY 28 AND 29, 1907

No. Card	Time	Steam	Vac.	M. P. Rec.	L. P. Rec.	Rev.	M.R.P.	I. H. P. H. P.	I. H. P. M. P.	I. H. P. L. P.	I. H. P. Total
1	3–25	170	26½	62	11	79	30.6	758.78	733.56	955.02	2447.36
2	3–40	170	26½	63	11	79	31.5	775.28	764.13	974.91	2514.32
3	4–35	170	26¼	63	12	79	30.4	742.98	733.56	955.02	2431.56
4	6–00	180	26	65	13½	81.5	32.1	822.50	788.31	1046.81	2657.62
5	7–40	172	26	60	11	79.5	31.5	763.57	768.96	1001.10	2533.63
6	10–00	180	25½	68	14	80.3	31.9	793.52	807.77	990.95	2592.24
7	11–00	178	25¼	70	15½	81	33.3	749.80	877.49	1102.40	2729.69
8	11–45	175	25¾	70	15	81	31.8	738.53	846.15	1019.99	2604.67

NOTE—Cards 1 to 5 inclusive taken January 28, 1907. Cards 6, 7, 8, taken with coal test, January 29, 1907.

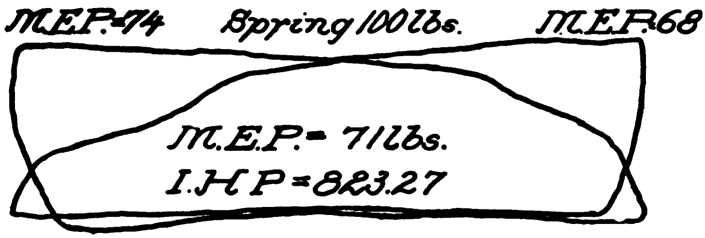
Coal used from 9.30 A. M. to 12.00 M., 14763 lbs.

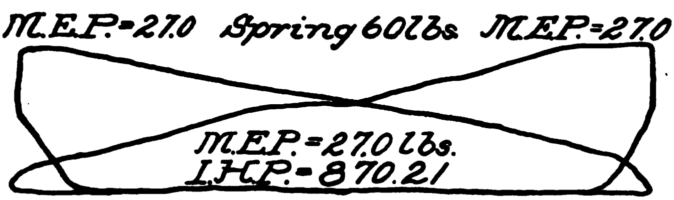
Coal used per hour, 5905 lbs.

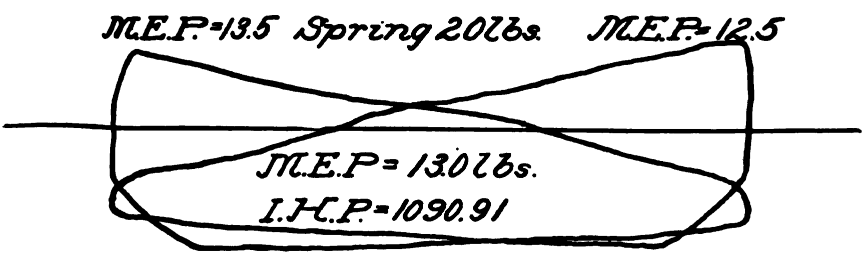
Coal used per I. H. P., per hour, 2.29 lbs.

Coal used per sq. foot of grate per hour, 20.9 lbs.

TUSCAN







By Pass Closed.

Time-10^{45} A.M. Date, 1-24-1907.

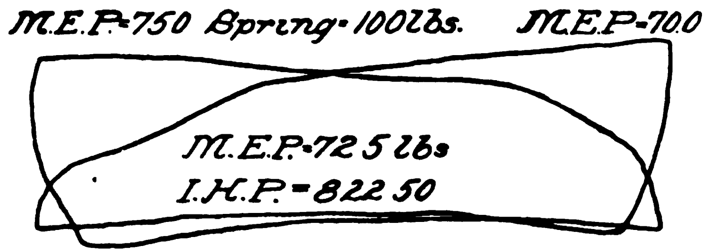
Card No. 1. Vac.-27.5

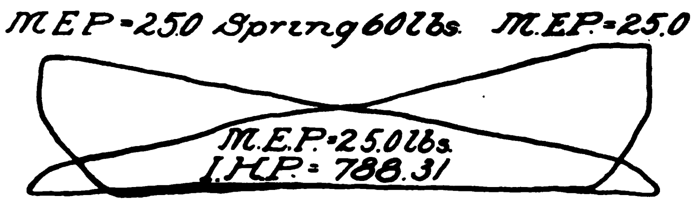
Steam-180 lbs. M.R.P.-330

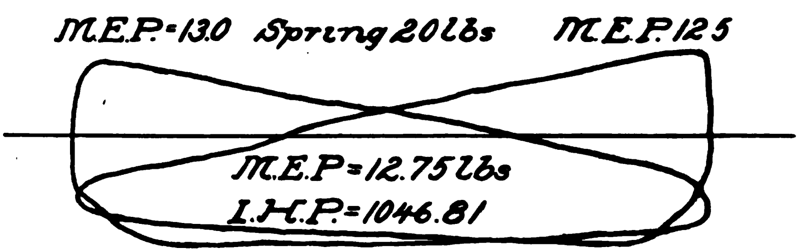
M.P. Rec. 66 L.P. Rec. 14.0

R.P.M.=83.3 I.H.P.=2784.39

TUSCAN



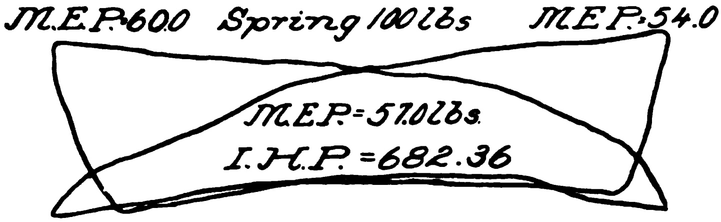


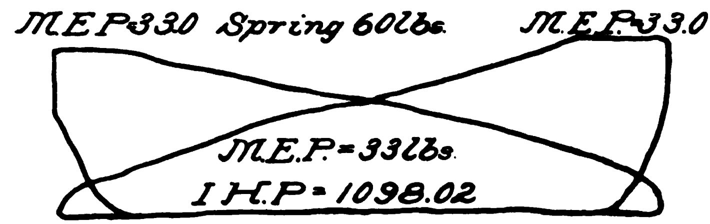


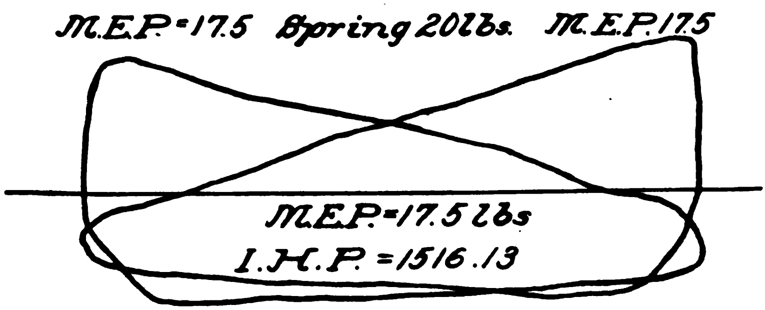
By Pass Closed.

Time-6^{00} P.M.	Date, 1-28-1907.
Card No. 4	Vac- 26.0
Steam- 180 lbs.	M.R.P. 32.1
M.P. Rec. 65	L.P. Rec. 13.5
R.P.M.- 81.5	I.H.P.=2657.62

TUSCAN







By Pass Open.

Time-2^{30} P.M.	Date-1-24-1907.
Card No. 13	Vac-27
Steam-190 lbs.	M.R.P. 37.9
M.P. Rec. 77.	L.P. Rec. 21.
R.P.M. - 86.	I.H.P. 3296.51.

The preceding series of diagrams are representative of modern marine engine practice. The data is sufficiently full to enable a thorough analysis to be made. They are worthy of close and careful study, and, being exact reproductions, can therefore be measured.

Further comment is unnecessary.

CHAPTER IV

Valve Diagrams

We will first describe the construction of the Zeuner diagram, and then the construction of the diagram for Marshall valve gear.

On plate 3 is shown valve diagrams for each cylinder of the engine shown on plate 1, the indicator diagrams of which are shown on page 95.

The construction will be made for top only (see plate 4), as the method for bottom is precisely the same.

Draw the horizontal line XX, and produce it to a sufficient length to take in the length of connecting rod between centers to same scale as selected for crank pin circle. Draw the vertical line YY, intersecting XX, in O. With O as center and radius equal to throw of crank, or half stroke, describe the crank pin circle A, B, C, D. This circle is drawn to any convenient scale; as shown it is drawn 3″=1 foot. Divide the diameter C, A, into 10 equal parts, each division representing $\frac{1}{10}$ of the stroke. With O as center, and radius equal to the eccentricity or half travel of the valve, describe the circle E, F, G, H. Now mark the end which is to be taken as top, and which one for bottom, selecting the right hand of diagram for top, as shown, and with G as center describe an arc i equal to the lead. It is better to make the valve diagram twice full size, as then the intersections of the different lines are shown with more distinctness.

Now set off from C a distance equal to the cut-off either in inches or percentage, and with a radius equal to connecting rod length as before, describe an arc, intersecting the crank pin circle in K. From O draw a diagonal line passing through K and cutting the circle of valve-travel in K_1. From K_1 draw a diagonal line tangent to the lead arc i and cutting the circle at L.

Through O draw a line OM perpendicular to K_1 L, cutting it in N. With O as center and ON as radius, describe an arc; ON is then the steam lap, and NM the maximum port opening.

Bisect the line OM, and with P as center describe the valve circles Q, R, S, and Q_1, R_1, S_1.

Through O draw a line parallel with K_1 L, and at the points of intersection with the travel circle T U_1 as centers describe arcs equal to the exhaust lap. If the exhaust lap is negative the circle will lie in the upper valve-circle, Q, R, S, and if positive it will lie in the lower valve-

circle Q_1, R_1, S_1. The reason for describing the arc at points T and U_1 is due to the fact that the intersection of the arc representing exhaust lap, with the valve-circle as at V, is rather difficult to exactly determine, and may cause variation.

From O draw diagonals tangent to the circles, and at the points where they cut the crank pin circle as at W, W_1, drop arcs with radius equal to radius of connecting rod, upon the diameter C, A. This gives the point of stroke at which release and compression takes place. With O as center and a radius equal to port opening plus exhaust lap describe an arc, cutting the lower valve-circle in Z, Z_1; from O draw diagonal lines through the points of intersection. This gives us the points between which the exhaust valve is full open.

Upon examining the diagram we see that the crank has to pass through the angle G, O, K_1, to arrive at the point where the steam is cut off; this point is shown at 1 where the lap-circle cuts the valve-circle.

Angle M, O, F, is the angle of advance. That is to say, when engine is turning over, the center of the eccentric sheave leads the center line of crank by 90 degrees plus the angle of advance; hence having the required lead, and point of cut-off we can by the construction determine the required angle.*

If the exhaust lap is negative, then the point of intersection of the lap circle with the valve-circle, point 2, shows where the valve opens to release the expanded steam. If, therefore, we desire to determine the point of release, we see that if it is desired to release later in the stroke the lap may have to be positive and if on the other hand we desire it earlier we need negative lap.

The distance between the intersection of the lap-circle with the diameter GE, and where the valve-circle cuts the diameter GE, is equal to the lead.

Again at point 3, where the lap-circle intersects the valve-circle this point of intersection shows where the valve starts to open for lead.

The analysis of the valve diagram enables us to determine the effects of any changes we may desire to make. Thus suppose we desire to cut off longer in the stroke, in other words to permit the steam to follow longer, the lead to remain unchanged. It is evident that to maintain the same lead, the steam lap must be reduced. Suppose, however, the lap is required to remain unchanged. It is evident that the lead must be reduced. The other changes involved will be left for the student to work out, and only by working out these different

* If engine turns under, the angle which the center of eccentric sheave makes with crank is 90 degrees—the angle of advance.

problems, in other words, constructing the diagram and discussing it, can he ever expect to be able to properly analyze it as it is impossible by mere reading to perform, and further, the subject is so broad and interesting that it is only by actual performance that one is able to grasp the details. There are several different diagrams used for analyzing the slide-valve operated by eccentrics, but the Zeuner is the most beautiful.

The diagrammatic work to the right of the diagram is only given to make the subject if possible more clear, and as before mentioned the diagrams shown on plate 3 should be very carefully studied.

The Marshall Valve Gear

The Marshall valve gear is one of the types of radial valve gears, which is used more extensively in marine practice than any other radial gear.

The diagram for Marshall valve gear and a valve diagram are shown on plate 5.

We will take a concrete case, and lay down the diagram, from the following data:

Travel of valve, $6\frac{13}{16}''$.
Lap of valve top, $1\frac{5}{16}''$.
Lap of valve bottom, $1\frac{1}{4}''$.
Lead top, $\frac{7}{16}''$.
Lead bottom, $\frac{1}{2}''$.
Maximum port opening, top $1\frac{1}{2}''$.
Maximum port opening, bottom $2\frac{3}{4}''$.
Cut-off top, 75.8 per cent. $=22\frac{3}{4}''$.
Cut-off bottom, 77.9 per cent. $=23\frac{3}{8}''$.
Stroke of piston $=30''$.
Eccentricity $=2\frac{1}{2}''$.
Length of stiff eccentric rod, 23.13''. .
Length of prolongation of eccentric rod, 16.03''.

Draw the horizontal line XX_1, and the vertical line YY_1, intersecting the horizontal line XX_1 in O.

Lay off a distance OC such that $OC=\sqrt{L^2-R^2}$, where L is the length of the stiff eccentric rod. OC in this case is given, namely, 23'', therefore, $L=\sqrt{OC^2+R^2}=23.13''$, and R is the eccentricity. From C lay-off a distance CD, and draw the vertical line UU_1.

With O as center and eccentricity as radius $2\frac{1}{2}''$ in this diagram, describe a circle, to any convenient scale. This diagram is drawn half size except where otherwise marked.

Now 5″ diameter circle drawn half size corresponds with 30″, the stroke of piston to a scale of 1″ = 1 foot. Therefore, with a scale of 1″ = 1 foot, set up on YY_1, produced, the stroke of engine as shown, and with a radius equal to the length of connecting rod between centers, in this case 5′ − 7½″, describe arcs cutting the circle in points 2, 4, 6, 8......30, etc., as shown. Now with C as center, and radius of length of radius rod, describe the arc A, B, in this case 12½″. With A and B as centers and the radius of 12½″ describe arcs E and F. With O_1, 2, 4, 6, 8, etc., as centers, and L as radius describe arcs on arc E, for one complete revolution in a head gear repeating the same process on arc F for astern gear. Now the distance CD is equal to the length of the prolongation of the stiff eccentric rod, "M." Therefore, from the points 0, 2, 4, 6, 8, etc., draw lines passing through the intersection of the arcs, on arc E and F as previously described. Measuring off from the points of intersection along the lines representing M, we get a series of points through which a fair curve is drawn, this elongated figure represents the oscillations of the point D, or the point of attachment of the valve-rod. The writer uses a beam compass with an extra attachment, placing needle point on points 0, 2, 4, 6, 8, etc., and the middle leg of compass on C, the other leg taken equal to the length of M; hence when arc is described on arc E, a corresponding arc is described at its proper distance, hence passing a line through the latter arc, a point is obtained; numbering these points as shown prevents confusion to one not accustomed to laying down the diagram, and until one is thoroughly acquainted with construction, it will pay to mark them; proceeding thus for one complete revolution we obtain points through which a fair curve is passed, giving us the elongated figure as shown.

Only the ahead motion has been considered. The astern motion is treated in precisely the same manner. If the student has not a beam compass handy, then a straight edge can be used, made as follows: Measure off the length L, and scrive marks upon the straight edge corresponding to the length O_1 C = L. Scrive a distance corresponding to CD_1 = M, therefore, the points of intersection can be accurately located. To the left of the diagram is drawn the stroke of piston to a scale of 1″ = 1 foot. This is divided into 15 equal parts representing 2 inch intervals of same.

The lap is laid off $1\frac{5}{16}$″ for top, 1¼″ for bottom.

With a pair of dividers the points for 2, 4, 6, 8, etc., of the elongated figure is laid off on the respective piston position. Connecting these points we obtain the figure as shown. Measuring the port opening for top we find 1½″ as required, for bottom we find 2¾″ as required. On

the diagram we lay off as shown, the lead, lap, and port opening. Observe that the point D_1 intersects the lead line for both top and bottom; this is as it should be, for when the crank is on top or bottom center the valve has opened for lead.

This engine is worked from the starboard side. If worked from the port side, the ahead position would be reversed, that is to say, ahead would be to the right and astern to the left of center line.

The eccentric coincides in this gear with the crank. The stiff eccentric-rod L is jointed at C to the radius rod AC, which swings on A. The gudgeon is attached to the radius arm, shown on plate 5, which is movable on fixed centers.

The prolongation M of the eccentric-rod L may form a slight angle with L if desirable. Conditions of design, however, control this.

It can be readily observed from diagram that the amount of lead is proportionate to the length M, and hence the term lead arm is frequently applied. The valve rod is joined at D_1 and the distance traversed represents the oscillations of the valve. The angle at which the radius-arm deviates on either side from the vertical through the fixed center is termed the deviation angle.

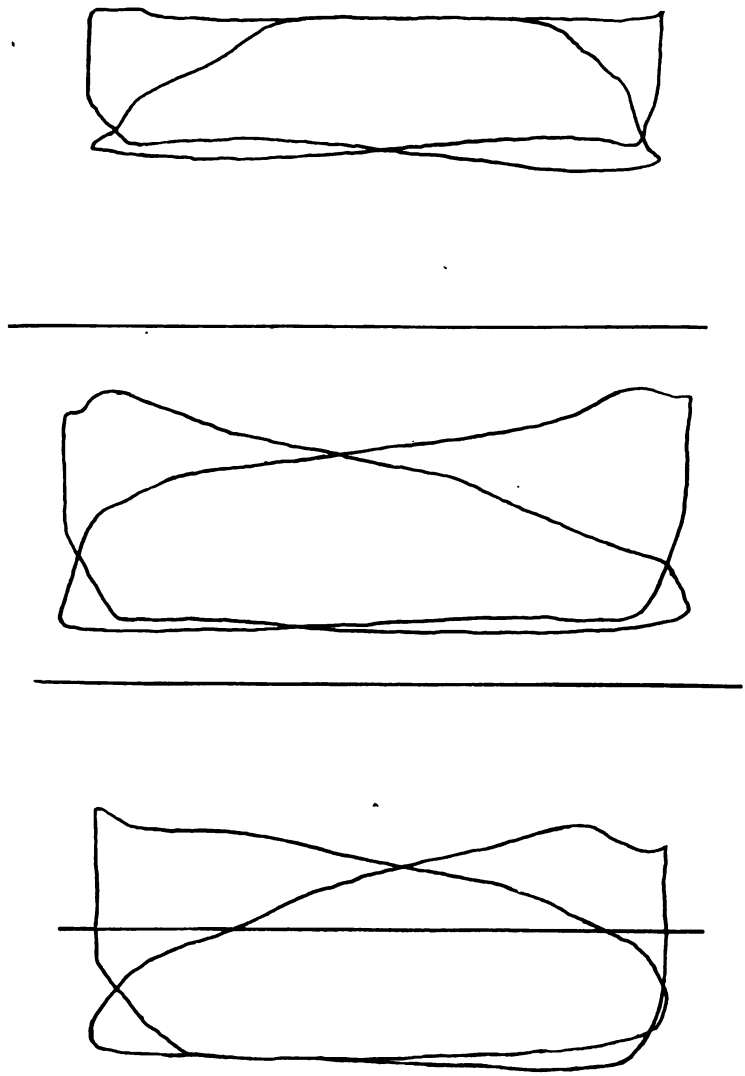
The crank-shaft revolves in the same direction in which the radius-arm deviates from the vertical.

As the center C travels through an arc described by the radius-rod AC the oscillations are greater above than below the center line, as will be noted. This difference between upper and lower oscillations has the following advantages:

The valve-openings are less for down stroke.
The cut-off is earlier.
The compression is earlier.
For the up stroke, the cut-off is later.
The valve-opening is greater.
The compression is later.
The momentum of the moving parts are, therefore, better balanced.

The difference between oscillations is effected by the length of radius-rod, and radius-arm.

The diagram has been marked to make its construction as clear as possible.



The set of diagrams shown above is from a triple expansion engine fitted with Marshall Valve Gear. These diagrams are fair types of those obtained with this gear, and same should be closely studied and compared with the other diagrams shown, as all other diagrams were taken from engines fitted with link-motion.

The publishing of peculiarly formed diagrams, showing various contours, has been purposely avoided, as it would be impossible to

show the very many forms of diagrams, and as it is only by a thorough grasp of the principles fundamental combined with practice that one can ever become proficient in analysis, it has been the author's aim to present these.

Plate 6 shows a section through the cylinders and valve chest of a triple expansion engine, and shows clearly the passages through which the steam travels from throttle valve to condenser. The H. P. and M. P. take steam on inside of valve and the L. P. on outside of valve. The receivers are cast with cylinders and are shown dotted. In this engine the H. P. crank leads.

It may be well to say in conclusion: Let the student take diagrams from either a compound or triple expansion engine with first H. P. crank leading, then if possible, diagrams from same type of engine with L. P. crank leading. Combine the diagrams, and note the difference under the various conditions. This way and this alone can he properly analyze.

If by writing this work I have been of help to those who are seeking this knowledge and who are willing to work hard for a clear understanding of this most interesting and vital subject, I shall feel amply repaid.

TABLE OF $\dfrac{1+\text{Hyp}\log r}{r}$

Let r = Rate of expansion.

$\dfrac{1}{r}$ = Cut-off.

r	$\frac{1}{r}$	$\frac{1+\text{Hyp}\log r}{r}$	r	$\frac{1}{r}$	$\frac{1+\text{Hyp}\log r}{r}$
1.33	0.752	0.9657	8.0	0.125	0.3849
1.4	0.714	0.9546	8.25	0.121	0.377
1.5	0.667	0.937	8.5	0.118	0.3694
1.6	0.625	0.9188	8.75	0.114	0.3622
1.7	0.588	0.9003	9.00	0.111	0.3552
1.75	0.571	0.8911	9.25	0.108	0.3486
1.8	0.556	0.882	9.5	0.105	0.3422
1.9	0.526	0.8641	9.75	0.103	0.3361
2.0	0.500	0.8465	10.00	0.100	0.3302
2.1	0.476	0.8294	10.25	0.097	0.3246
2.2	0.455	0.8129	10.50	0.095	0.3191
2.25	0.444	0.8048	10.75	0.093	0.315
2.75	0.364	0.7315	11.00	0.091	0.3088
3.00	0.333	0.6995	11.25	0.089	0.304
3.25	0.308	0.6703	11.50	0.087	0.2994
3.75	0.267	0.6191	11.75	0.0851	0.2947
4.0	0.25	0.5965	12.00	0.0833	0.2904
4.25	0.235	0.5757	12.25	0.0816	0.2861
4.5	0.222	0.5564	12.5	0.08	0.2821
5.0	0.200	0.5219	12.75	0.0784	0.2781
5.25	0.190	0.5063	13.	0.0769	0.2741
5.5	0.182	0.4917	13.25	0.0755	0.2705
5.75	0.174	0.4781	13.5	0.0741	0.2668
6.	0.167	θ.4652	13.75	0.0727	0.2633
6.25	0.160	0.4532	14.	0.0714	0.2599
6.5	0.154	0.4418	15.	0.0667	0.2472
6.75	0.148	0.431	16.	0.0625	0.2358
7.0	0.143	0.4208	17.	0.0588	0.2255
7.25	0.138	0.4111	18.	0.055	0.2161
7.5	0.133	0.4019	20.	0.050	0.1998
7.75	0.129	0.3932			

TABLE

CONTAINING THE

COMMON LOGARITHMS OF NUMBERS

FROM 1 TO 10,000

To obtain the hyperbolic logarithm of a number
multiply the common logarithm of
the number by 2.302585

N.	0	1	2	3	4	5	6	7	8	9	D.
100	00 0000	00 0434	00 0868	00 1301	00 1734	00 2166	00 2598	00 3029	00 3461	00 3891	432
101	4321	4751	5181	5609	6038	6466	6894	7321	7748	8174	428
102	8600	9026	9451	9876	01 0300	01 0724	01 1147	01 1570	01 1993	01 2415	424
103	01 2837	01 3259	01 3680	01 4100	4521	4940	5360	5779	6197	6616	420
104	7033	7451	7868	8284	8700	9116	9532	9947	02 0361	02 0775	416
105	02 1189	02 1603	02 2016	02 2428	02 2841	02 3252	02 3664	02 4075	02 4486	02 4896	412
106	5306	5715	6125	6533	6942	7350	7757	8164	8571	8978	408
107	9384	9789	03 0195	03 0600	03 1004	03 1408	03 1812	03 2216	03 2619	03 3021	404
108	03 3424	03 3826	4227	4628	5029	5430	5830	6230	6629	7028	400
109	7426	7825	8223	8620	9017	9414	9811	04 0207	04 0602	04 0998	397
110	04 1393	04 1787	04 2182	04 2576	04 2969	04 3362	04 3755	04 4148	04 4540	04 4932	393
111	5323	5714	6105	6495	6885	7275	7664	8053	8442	8830	390
112	9218	9606	9993	05 0380	05 0766	05 1153	05 1538	05 1924	05 2309	05 2694	386
113	05 3078	05 3463	05 3846	4230	4613	4996	5378	5760	6142	6524	383
114	6905	7286	7666	8046	8426	8805	9185	9563	9942	06 0320	379
115	06 0698	06 1075	06 1452	06 1829	06 2206	06 2582	06 2958	06 3333	06 3709	06 4083	376
116	4458	4832	5206	5580	5953	6326	6699	7071	7443	7815	373
117	8186	8557	8928	9298	9668	07 0038	07 0407	07 0776	07 1145	07 1514	370
118	07 1882	07 2250	07 2617	07 2985	07 3352	3718	4085	4451	4816	5182	366
119	5547	5912	6276	6640	7004	7368	7731	8094	8457	8819	363
120	07 9181	07 9543	07 9904	08 0266	08 0626	08 0987	08 1347	08 1707	08 2067	08 2426	360
121	08 2785	08 3144	08 3503	3861	4219	4576	4934	5291	5647	6004	357
122	6360	6716	7071	7426	7781	8136	8490	8845	9198	9552	355
123	9905	09 0258	09 0611	09 0963	09 1315	09 1667	09 2018	09 2370	09 2721	09 3071	352
124	09 3422	3772	4122	4471	4820	5169	5518	5866	6215	6562	349
125	09 6910	09 7257	09 7604	09 7951	09 8298	09 8644	09 8990	09 9335	09 9681	10 0026	346
126	10 0371	10 0715	10 1059	10 1403	10 1747	10 2091	10 2434	10 2777	10 3119	3462	343
127	3804	4146	4487	4828	5169	5510	5851	6191	6531	6871	341
128	7210	7549	7888	8227	8565	8903	9241	9579	9916	11 0253	338
129	11 0590	11 0926	11 1263	11 1599	11 1934	11 2270	11 2605	11 2940	11 3275	3609	335
130	11 3943	11 4277	11 4611	11 4944	11 5278	11 5611	11 5943	11 6276	11 6608	11 6940	333
131	7271	7603	7934	8265	8595	8926	9256	9586	9915	12 0245	330
132	12 0574	12 0903	12 1231	12 1560	12 1888	12 2216	12 2544	12 2871	12 3198	3525	328
133	3852	4178	4504	4830	5156	5481	5806	6131	6456	6781	325
134	7105	7429	7753	8076	8399	8722	9045	9368	9690	13 0012	323
135	13 0334	13 0655	13 0977	13 1298	13 1619	13 1939	13 2260	13 2580	13 2900	13 3219	321
136	3539	3858	4177	4496	4814	5133	5451	5769	6086	6403	318
137	6721	7037	7354	7671	7987	8303	8618	8934	9249	9564	316
138	9879	14 0194	14 0508	14 0822	14 1136	14 1450	14 1763	14 2076	14 2389	14 2702	314
139	14 3015	3327	3639	3951	4263	4574	4885	5196	5507	5818	311
140	14 6128	14 6438	14 6748	14 7058	14 7367	14 7676	14 7985	14 8294	14 8603	14 8911	309
141	9219	9527	9835	15 0142	15 0449	15 0756	15 1063	15 1370	15 1676	15 1982	307
142	15 2288	15 2594	15 2900	3205	3510	3815	4120	4424	4728	5032	305
143	5336	5040	5943	6246	6549	6852	7154	7457	7759	8061	303
144	8362	8664	8965	9266	9567	9868	16 0168	16 0469	16 0769	16 1068	301
145	16 1368	16 1667	16 1967	16 2266	16 2564	16 2863	16 3161	16 3460	16 3758	16 4055	299
146	4353	4650	4947	5244	5541	5838	6134	6430	6726	7022	297
147	7317	7613	7908	8203	8497	8792	9086	9380	9674	9968	295
148	17 0262	17 0555	17 0848	17 1141	17 1434	17 1726	17 2019	17 2311	17 2603	17 2895	293
149	3186	3478	3769	4060	4351	4641	4932	5222	5512	5802	291
150	17 6091	17 6381	17 6670	17 6959	17 7248	17 7536	17 7825	17 8113	17 8401	17 8689	289
151	8977	9264	9552	9839	18 0126	18 0413	18 0699	18 0986	18 1272	18 1558	287
152	18 1844	18 2129	18 2415	18 2700	2985	3270	3555	3839	4123	4407	285
153	4691	4975	5259	5542	5825	6108	6391	6674	6956	7239	283
154	7521	7803	8084	8366	8647	8928	9209	9490	9771	19 0051	281
155	19 0332	19 0612	19 0892	19 1171	19 1451	19 1730	19 2010	19 2289	19 2567	19 2846	279
156	3125	3403	3681	3959	4237	4514	4792	5069	5346	5623	278
157	5900	6176	6453	6729	7005	7281	7556	7832	8107	8382	276
158	8657	8932	9206	9481	9755	20 0029	20 0303	20 0577	20 0850	20 1124	274
159	20 1397	20 1670	20 1943	20 2216	20 2488	2761	3033	3305	3577	3848	272
N.	0	1	2	3	4	5	6	7	8	9	D.

N.	0	1	2	3	4	5	6	7	8	9	D.
160	20 4120	20 4391	20 4663	20 4934	20 5204	20 5475	20 5746	20 6016	20 6286	20 6556	271
161	6826	7096	7365	7634	7904	8173	8441	8710	8979	9247	269
162	9515	9783	21 0051	21 0319	21 0586	21 0853	21 1121	21 1388	21 1654	21 1921	267
163	21 2188	21 2454	2720	2986	3252	3518	3783	4049	4314	4579	266
164	4844	5109	5373	5638	5902	6166	6430	6694	6957	7221	264
165	21 7484	21 7747	21 8010	21 8273	21 8536	21 8798	21 9060	21 9323	21 9585	21 9846	262
166	22 0108	22 0370	22 0631	22 0892	22 1153	22 1414	22 1675	22 1936	22 2196	22 2456	261
167	2716	2976	3236	3496	3755	4015	4274	4533	4792	5051	259
168	5309	5568	5826	6084	6342	6600	6858	7115	7372	7630	258
169	7887	8144	8400	8657	8913	9170	9426	9682	9938	23 0193	256
170	23 0449	23 0704	23 0960	23 1215	23 1470	23 1724	23 1979	23 2234	23 2488	23 2742	255
171	2996	3250	3504	3757	4011	4264	4517	4770	5023	5276	253
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463	5581	5675	5769	5862	5956	6050	6143	6237	6331	6424	94
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466	8386	8479	8572	8665	8759	8852	8945	9038	9131	9224	93
467	9317	9410	9503	9596	9689	9782	9875	9967	67 0060	67 0153	93
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479	68 0336	68 0426	68 0517	68 0607	68 0698	68 0789	68 0879	0970	1060	1151	91
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550	74 0363	74 0442	74 0521	74 0600	74 0678	74 0757	74 0836	74 0915	74 0994	74 1073	79
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589	77 0115	77 0189	77 0263	77 0336	77 0410	77 0484	77 0557	77 0631	77 0705	0778	74
590	77 0852	77 0926	77 0999	77 1073	77 1146	77 1220	77 1293	77 1367	77 1440	77 1514	74
591	1587	1661	1734	1808	1881	1955	2028	2102	2175	2248	73
592	2322	2395	2468	2542	2615	2688	2762	2835	2908	2981	73
593	3055	3128	3201	3274	3348	3421	3494	3567	3640	3713	73
594	3786	3860	3933	4006	4079	4152	4225	4298	4371	4444	73
595	77 4517	77 4590	77 4663	77 4736	77 4809	77 4882	77 4955	77 5028	77 5100	77 5173	73
596	5246	5319	5392	5465	5538	5610	5683	5756	5829	5902	73
597	5974	6047	6120	6193	6265	6338	6411	6483	6556	6629	73
598	6701	6774	6846	6919	6992	7064	7137	7209	7282	7354	73
599	7427	7499	7572	7644	7717	7789	7862	7934	8006	8079	72
600	77 8151	77 8224	77 8296	77 8368	77 8441	77 8513	77 8585	77 8658	77 8730	77 8802	72
601	8874	8947	9019	9091	9163	9236	9308	9380	9452	9524	72
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603	78 0317	78 0389	78 0461	78 0533	78 0605	78 0677	0749	0821	0893	0965	72
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605	78 1755	78 1827	78 1899	78 1971	78 2042	78 2114	78 2186	78 2258	78 2329	78 2401	72
606	2473	2544	2616	2688	2759	2831	2902	2974	3046	3117	72
607	3189	3260	3332	3403	3475	3546	3618	3689	3761	3832	71
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610	78 5330	78 5401	78 5472	78 5543	78 5615	78 5686	78 5757	78 5828	78 5899	78 5970	71
611	6041	6112	6183	6254	6325	6396	6467	6538	6609	6680	71
612	6751	6822	6893	6964	7035	7106	7177	7248	7319	7390	71
613	7460	7531	7602	7673	7744	7815	7885	7956	8027	8098	71
614	8168	8239	8310	8381	8451	8522	8593	8663	8734	8804	71
615	78 8875	78 8946	78 9016	78 9087	78 9157	78 9228	78 9299	78 9369	78 9440	78 9510	71
616	9581	9651	9722	9792	9863	9933	79 0004	79 0074	79 0144	79 0215	70
617	79 0285	79 0356	79 0426	79 0496	79 0567	79 0637	0707	0778	0848	0918	70
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620	79 2392	79 2462	79 2532	79 2602	79 2672	79 2742	79 2812	79 2882	79 2952	79 3022	70
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625	79 5880	79 5949	79 6019	79 6088	79 6158	79 6227	79 6297	79 6366	79 6436	79 6505	69
626	6574	6644	6713	6782	6852	6921	6990	7060	7129	7198	69
627	7268	7337	7406	7475	7545	7614	7683	7752	7821	7890	69
628	7960	8029	8098	8167	8236	8305	8374	8443	8513	8582	69
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631	80 0029	80 0098	80 0167	80 0236	80 0305	80 0373	80 0442	80 0511	80 0580	80 0648	69
632	0717	0786	0854	0923	0992	1061	1129	1198	1266	1335	69
633	1404	1472	1541	1609	1678	1747	1815	1884	1952	2021	69
634	2089	2158	2226	2295	2363	2432	2500	2568	2637	2705	68
635	80 2774	80 2842	80 2910	80 2979	80 3047	80 3116	80 3184	80 3252	80 3321	80 3389	68
636	3457	3525	3594	3662	3730	3798	3867	3935	4003	4071	68
637	4139	4208	4276	4344	4412	4480	4548	4616	4685	4753	68
638	4821	4889	4957	5025	5093	5161	5229	5297	5365	5433	68
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642	7535	7603	7670	7738	7806	7873	7941	8008	8076	8143	68
643	8211	8279	8346	8414	8481	8549	8616	8684	8751	8818	67
644	8886	8953	9021	9088	9156	9223	9290	9358	9425	9492	67
645	80 9560	80 9627	80 9694	80 9762	80 9829	80 9896	80 9964	81 0031	81 0098	81 0165	67
646	81 0233	81 0300	81 0367	81 0434	81 0501	81 0569	81 0636	0703	0770	0837	67
647	0904	0971	1039	1106	1173	1240	1307	1374	1441	1508	67
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651	3581	3648	3714	3781	3848	3914	3981	4048	4114	4181	67
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653	4913	4980	5046	5113	5179	5246	5312	5378	5445	5511	66
654	5578	5644	5711	5777	5843	5910	5976	6042	6109	6175	66
655	81 6241	81 6308	81 6374	81 6440	81 6506	81 6573	81 6639	81 6705	81 6771	81 6838	66
656	6904	6970	7036	7102	7169	7235	7301	7367	7433	7499	66
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658	8226	8292	8358	8424	8490	8556	8622	8688	8754	8820	66
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661	82 0201	82 0267	82 0333	82 0399	82 0464	82 0530	82 0595	0661	0727	0792	66
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670	82 6075	82 6140	82 6204	82 6269	82 6334	82 6399	82 6464	82 6528	82 6593	82 6658	65
671	6723	6787	6852	6917	6981	7046	7111	7175	7240	7305	65
672	7369	7434	7499	7563	7628	7692	7757	7821	7886	7951	65
673	8015	8080	8144	8209	8273	8338	8402	8467	8531	8595	64
674	8660	8724	8789	8853	8918	8982	9046	9111	9175	9239	64
675	82 9304	82 9368	82 9432	82 9497	82 9561	82 9625	82 9690	82 9754	82 9818	82 9882	64
676	9947	83 0011	83 0075	83 0139	83 0204	83 0268	83 0332	83 0396	83 0460	83 0525	64
677	83 0589	0653	0717	0781	0845	0909	0973	1037	1102	1166	64
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679	1870	1934	1998	2062	2126	2189	2253	2317	2381	2445	64
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685	83 5691	83 5754	83 5817	83 5881	83 5944	83 6007	83 6071	83 6134	83 6197	83 6261	63
686	6324	6387	6451	6514	6577	6641	6704	6767	6830	6894	63
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691	9478	9541	9604	9667	9729	9792	9855	9918	9981	84 0043	63
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694	1359	1422	1485	1547	1610	1672	1735	1797	1860	1922	63
695	84 1985	84 2047	84 2110	84 2172	84 2235	84 2297	84 2360	84 2422	84 2484	84 2547	62
696	2609	2672	2734	2796	2859	2921	2983	3046	3108	3170	62
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698	3855	3918	3980	4042	4104	4166	4229	4291	4353	4415	62
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701	5718	5780	5842	5904	5966	6028	6090	6151	6213	6275	62
702	6337	6399	6461	6523	6585	6646	6708	6770	6832	6894	62
703	6955	7017	7079	7141	7202	7264	7326	7388	7449	7511	62
704	7573	7634	7696	7758	7819	7881	7943	8004	8066	8128	62
705	84 8189	84 8251	84 8312	84 8374	84 8435	84 8497	84 8559	84 8620	84 8682	84 8743	62
706	8805	8866	8928	8989	9051	9112	9174	9235	9297	9358	61
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708	85 0033	85 0095	85 0156	85 0217	85 0279	85 0340	85 0401	85 0462	85 0524	85 0585	61
709	0646	0707	0769	0830	0891	0952	1014	1075	1136	1197	61
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711	1870	1931	1992	2053	2114	2175	2236	2297	2358	2419	61
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715	85 4306	85 4367	85 4428	85 4488	85 4549	85 4610	85 4670	85 4731	85 4792	85 4852	61
716	4913	4974	5034	5095	5156	5216	5277	5337	5398	5459	61
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721	7935	7995	8056	8116	8176	8236	8297	8357	8417	8477	60
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729	2728	2787	2847	2906	2966	3025	3085	3144	3204	3263	60
730	86 3323	86 3382	86 3442	86 3501	68 3561	86 3620	86 3680	86 3739	86 3799	86 3858	59
731	3917	3977	4036	4096	4155	4214	4274	4333	4392	4452	59
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734	5696	5755	5814	5874	5933	5992	6051	6110	6169	6228	59
735	86 6287	86 6346	86 6405	86 6465	86 6524	86 6583	86 6642	86 6701	86 6760	86 6819	59
736	6878	6937	6996	7055	7114	7173	7232	7291	7350	7409	59
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742	87 0404	87 0462	87 0521	87 0579	0638	0696	0755	0813	0872	0930	58
743	0989	1047	1106	1164	1223	1281	1339	1398	1456	1515	58
744	1573	1631	1690	1748	1806	1865	1923	1981	2040	2098	58
745	87 2156	87 2215	87 2273	87 2331	87 2389	87 2448	87 2506	87 2564	87 2622	87 2681	58
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753	6795	6853	6910	6968	7026	7083	7141	7199	7256	7314	58
754	7371	7429	7487	7544	7602	7659	7717	7774	7832	7889	58
755	87 7947	87 8004	87 8062	87 8119	87 8177	87 8234	87 8292	87 8349	87 8407	87 8464	57
756	8522	8579	8637	8694	8752	8809	8866	8924	8981	9039	57
757	9096	9153	9211	9268	9325	9383	9440	9497	9555	9612	57
758	9669	9726	9784	9841	9898	9956	88 0013	88 0070	88 0127	88 0185	57
759	88 0242	88 0299	88 0356	88 0413	88 0471	88 0528	0585	0642	0699	0756	57
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762	1955	2012	2069	2126	2183	2240	2297	2354	2411	2468	57
763	2525	2581	2638	2695	2752	2809	2866	2923	2980	3037	57
764	3093	3150	3207	3264	3321	3377	3434	3491	3548	3605	57
765	88 3661	88 3718	88 3775	88 3832	88 3888	88 3945	88 4002	88 4059	88 4115	88 4172	57
766	4229	4285	4342	4399	4455	4512	4569	4625	4682	4739	57
767	4795	4852	4909	4965	5022	5078	5135	5192	5248	5305	57
768	5361	5418	5474	5531	5587	5644	5700	5757	5813	5870	57
769	5926	5983	6039	6096	6152	6209	6265	6321	6378	6434	56
770	88 6491	88 6547	88 6604	88 6660	88 6716	88 6773	88 6829	88 6885	88 6942	88 6998	56
771	7054	7111	7167	7223	7280	7336	7392	7449	7505	7561	56
772	7617	7674	7730	7786	7842	7898	7955	8011	8067	8123	56
773	8179	8236	8292	8348	8404	8460	8516	8573	8629	8685	56
774	8741	8797	8853	8909	8965	9021	9077	9134	9190	9246	56
775	88 9302	88 9358	88 9414	88 9470	88 9526	88 9582	88 9638	88 9694	88 9750	88 9806	56
776	9862	9918	9974	89 0030	89 0086	89 0141	89 0197	89 0253	89 0309	89 0365	56
777	89 0421	89 0477	89 0533	0589	0645	0700	0756	0812	0868	0924	56
778	0980	1035	1091	1147	1203	1259	1314	1370	1426	1482	56
779	1537	1593	1649	1705	1760	1816	1872	1928	1983	2039	56
780	89 2095	89 2150	89 2206	89 2262	89 2317	89 2373	89 2429	89 2484	89 2540	89 2595	56
781	2651	2707	2762	2818	2873	2929	2985	3040	3096	3151	56
782	3207	3262	3318	3373	3429	3484	3540	3595	3651	3706	56
783	3762	3817	3873	3928	3984	4039	4094	4150	4205	4261	55
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793	9273	9328	9383	9437	9492	9547	9602	9656	9711	9766	55
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795	90 0367	90 0422	90 0476	90 0531	90 0586	90 0640	90 0695	90 0749	90 0804	90 0859	55
796	0913	0968	1022	1077	1131	1186	1240	1295	1349	1404	55
797	1458	1513	1567	1622	1676	1731	1785	1840	1894	1948	54
798	2003	2057	2112	2166	2221	2275	2329	2384	2438	2492	54
799	2547	2601	2655	2710	2764	2818	2873	2927	2981	3036	54
800	90 3090	90 3144	90 3199	90 3253	90 3307	90 3361	90 3416	90 3470	90 3524	90 3578	54
801	3633	3687	3741	3795	3849	3904	3958	4012	4066	4120	54
802	4174	4229	4283	4337	4391	4445	4499	4553	4607	4661	54
803	4716	4770	4824	4878	4932	4986	5040	5094	5148	5202	54
804	5256	5310	5364	5418	5472	5526	5580	5634	5688	5742	54
805	90 5796	90 5850	90 5904	90 5958	90 6012	90 6066	90 6119	90 6173	90 6227	90 6281	54
806	6335	6389	6443	6497	6551	6604	6658	6712	6766	6820	54
807	6874	6927	6981	7035	7089	7143	7196	7250	7304	7358	54
808	7411	7465	7519	7573	7626	7680	7734	7787	7841	7895	54
809	7949	8002	8056	8110	8163	8217	8270	8324	8378	8431	54
810	90 8485	90 8539	90 8592	90 8646	90 8699	90 8753	90 8807	90 8860	90 8914	90 8967	54
811	9021	9074	9128	9181	9235	9289	9342	9396	9449	9503	54
812	9556	9610	9663	9716	9770	9823	9877	9930	9984	91 0037	53
813	91 0091	91 0144	91 0197	91 0251	91 0304	91 0358	91 0411	91 0464	91 0518	0571	53
814	0624	0678	0731	0784	0838	0891	0944	0998	1051	1104	53
815	91 1158	91 1211	91 1264	91 1317	91 1371	91 1424	91 1477	91 1530	91 1584	91 1637	53
816	1690	1743	1797	1850	1903	1956	2009	0263	2116	2169	53
817	2222	2275	2328	2381	2435	2488	2541	2594	2647	2700	53
818	2753	2806	2859	2913	2966	3019	3072	3125	3178	3231	53
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821	4343	4396	4449	4502	4555	4608	4660	4713	4766	4819	53
822	4872	4925	4977	5030	5083	5136	5189	5241	5294	5347	53
823	5400	5453	5505	5558	5611	5664	5716	5769	5822	5875	53
824	5927	5980	6033	6085	6138	6191	6243	6296	6349	6401	53
825	91 6454	91 6507	91 6559	91 6612	91 6664	91 6717	91 6770	91 6822	91 6875	91 6927	53
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827	7506	7558	7611	7663	7716	7768	7820	7873	7925	7978	52
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829	8555	8607	8659	8712	8764	8816	8869	8921	8973	9026	52
830	91 9078	91 9130	91 9183	91 9235	91 9287	91 9340	91 9392	91 9444	91 9496	91 9549	52
831	9601	9653	9706	9758	9810	9862	9914	9967	92 0019	92 0071	52
832	92 0123	92 0176	92 0228	92 0280	92 0332	92 0384	92 0436	92 0489	0541	0593	52
833	0645	0697	0749	0801	0853	0906	0958	1010	1062	1114	52
834	1166	1218	1270	1322	1374	1426	1478	1530	1582	1634	52
835	92 1686	92 1738	92 1790	92 1842	92 1894	92 1946	92 1998	92 2050	92 2102	92 2154	52
836	2206	2258	2310	2362	2414	2466	2518	2570	2622	2674	52
837	2725	2777	2829	2881	2933	2985	3037	3089	3140	3192	52
838	3244	3296	3348	3399	3451	3503	3555	3607	3658	3710	52
839	3762	3814	3865	3917	3969	4021	4072	4124	4176	4228	52
840	92 4279	92 4331	92 4383	92 4434	92 4486	92 4538	92 4589	92 4641	92 4693	92 4744	52
841	4796	4848	4899	4951	5003	5054	5106	5157	5209	5261	52
842	5312	5364	5415	5467	5518	5570	5621	5673	5725	5776	52
843	5828	5879	5931	5982	6034	6085	6137	6188	6240	6291	51
844	6342	6394	6445	6497	6548	6600	6651	6702	6754	6805	51
845	92 6857	92 6908	92 6959	92 7011	92 7062	92 7114	92 7165	92 7216	92 7268	92 7319	51
846	7370	7422	7473	7524	7576	7627	7678	7730	7781	7832	51
847	7883	7935	7986	8037	8088	8140	8191	8242	8293	8345	51
848	8396	8447	8498	8549	8601	8652	8703	8754	8805	8857	51
849	8908	8959	9010	9061	9112	9163	9215	9266	9317	9368	51
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851	9930	9981	93 0032	93 0083	93 0134	93 0185	93 0236	93 0287	93 0338	93 0389	51
852	93 0440	93 0491	0542	0592	0643	0694	0745	0796	0847	0898	51
853	0949	1000	1051	1102	1153	1204	1254	1305	1356	1407	51
854	1458	1509	1560	1610	1661	1712	1763	1814	1865	1915	51
855	93 1966	93 2017	93 2068	93 2118	93 2169	93 2220	93 2271	93 2322	93 2372	93 2423	51
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857	2981	3031	3082	3133	3183	3234	3285	3335	3386	3437	51
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859	3993	4044	4094	4145	4195	4246	4296	4347	4397	4448	51
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861	5003	5054	5104	5154	5205	5255	5306	5356	5406	5457	50
862	5507	5558	5608	5658	5709	5759	5809	5860	5910	5960	50
863	6011	6061	6111	6162	6212	6262	6313	6363	6413	6463	50
864	6514	6564	6614	6665	6715	6765	6815	6865	6916	6966	50
865	93 7016	93 7066	93 7117	93 7167	93 7217	93 7267	93 7317	93 7367	93 7418	93 7468	50
866	7518	7568	7618	7668	7718	7769	7819	7869	7919	7969	50
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872	0516	0566	0616	0666	0716	0765	0815	0865	0915	0964	50
873	1014	1064	1114	1163	1213	1263	1313	1362	1412	1462	50
874	1511	1561	1611	1660	1710	1760	1809	1859	1909	1958	50
875	94 2008	94 2058	94 2107	94 2157	94 2207	94 2256	94 2306	94 2355	94 2405	94 2455	50
876	2504	2554	2603	2653	2702	2752	2801	2851	2901	2950	50
877	3000	3049	3099	3148	3198	3247	3297	3346	3396	3445	49
878	3495	3544	3593	3643	3692	3742	3791	3841	3890	3939	49
879	3989	4038	4088	4137	4186	4236	4285	4335	4384	4433	49
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881	4976	5025	5074	5124	5173	5222	5272	5321	5370	5419	49
882	5469	5518	5567	5616	5665	5715	5764	5813	5862	5912	49
883	5961	6010	6059	6108	6157	6207	6256	6305	6354	6403	49
884	6452	6501	6551	6600	6649	6698	6747	6796	6845	6894	49
885	94 6943	94 6992	94 7041	94 7090	94 7140	94 7189	94 7238	94 7287	94 7336	94 7385	49
886	7434	7483	7532	7581	7630	7679	7728	7777	7826	7875	49
887	7924	7973	8022	8070	8119	8168	8217	8266	8315	8364	49
888	8413	8462	8511	8560	8609	8657	8706	8755	8804	8853	49
889	8902	8951	8999	9048	9097	9146	9195	9244	9292	9341	49
890	94 9390	94 9439	94 9488	94 9536	94 9585	94 9634	94 9683	94 9731	94 9780	94 9829	49
891	9878	9926	9975	95 0024	95 0073	95 0121	95 0170	95 0219	95 0267	95 0316	49
892	95 0365	95 0414	95 0462	0511	0560	0608	0657	0706	0754	0803	49
893	0851	0900	0949	0997	1046	1095	1143	1192	1240	1289	49
894	1338	1386	1435	1483	1532	1580	1629	1677	1726	1775	49
895	95 1823	95 1872	95 1920	95 1969	95 2017	95 2066	95 2114	95 2163	95 2211	95 2260	48
896	2308	2356	2405	2453	2502	2550	2599	2647	2696	2744	48
897	2792	2841	2889	2938	2986	3034	3083	3131	3180	3228	48
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900	95 4243	95 4291	95 4339	95 4387	95 4435	95 4484	95 4532	95 4580	95 4628	95 4677	48
901	4725	4773	4821	4869	4918	4966	5014	5062	5110	5158	48
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903	5688	5736	5784	5832	5880	5928	5976	6024	6072	6120	48
904	6168	6216	6265	6313	6361	6409	6457	6505	6553	6601	48
905	95 6649	95 6697	95 6745	95 6793	95 6840	95 6888	95 6936	95 6984	95 7032	95 7080	48
906	7128	7176	7224	7272	7320	7368	7416	7464	7512	7559	48
907	7607	7655	7703	7751	7799	7847	7894	7942	7990	8038	48
908	8086	8134	8181	8229	8277	8325	8373	8421	8468	8516	48
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910	95 9041	95 9089	95 9137	95 9185	95 9232	95 9280	95 9328	95 9375	95 9423	95 9471	48
911	9518	9566	9614	9661	9709	9757	9804	9852	9900	9947	48
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915	96 1421	96 1469	96 1516	96 1563	96 1611	96 1658	96 1706	96 1753	96 1801	96 1848	47
916	1895	1943	1990	2038	2085	2132	2180	2227	2275	2322	47
917	2369	2417	2464	2511	2559	2606	2653	2701	2748	2795	47
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924	5672	5719	5766	5813	5860	5907	5954	6001	6048	6095	47
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927	7080	7127	7173	7220	7267	7314	7361	7408	7454	7501	47
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934	97 0347	97 0393	97 0440	0486	0533	0579	0626	0672	0719	0765	46
935	97 0812	97 0858	97 0904	97 0951	97 0997	97 1044	97 1090	97 1137	97 1183	97 1229	46
936	1276	1322	1369	1415	1461	1508	1554	1601	1647	1693	46
937	1740	1786	1832	1879	1925	1971	2018	2064	2110	2157	46
938	2203	2249	2295	2342	2388	2434	2481	2527	2573	2619	46
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941	3590	3636	3682	3728	3774	3820	3866	3913	3959	4005	46
942	4051	4097	4143	4189	4235	4281	4327	4374	4420	4466	46
943	4512	4558	4604	4650	4696	4742	4788	4834	4880	4926	46
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945	97 5432	97 5478	97 5524	97 5570	97 5616	97 5662	97 5707	97 5753	97 5799	97 5845	46
946	5891	5937	5983	6029	6075	6121	6167	6212	6258	6304	46
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948	6808	6854	6900	6946	6992	7037	7083	7129	7175	7220	46
949	7266	7312	7358	7403	7449	7495	7541	7586	7632	7678	46
950	97 7724	97 7769	97 7815	97 7861	97 7906	97 7952	97 7998	97 8043	97 8089	97 8135	46
951	8181	8226	8272	8317	8363	8409	8454	8500	8546	8591	46
952	8637	8683	8728	8774	8819	8865	8911	8956	9002	9047	46
953	9093	9138	9184	9230	9275	9321	9366	9412	9457	9503	46
954	9548	9594	9639	9685	9730	9776	9821	9867	9912	9958	46
955	98 0003	98 0049	98 0094	98 0140	98 0185	98 0231	98 0276	98 0322	98 0367	98 0412	45
956	0458	0503	0549	0594	0640	0685	0730	0776	0821	0867	45
957	0912	0957	1003	1048	1093	1139	1184	1229	1275	1320	45
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961	2723	2769	2814	2859	2904	2949	2994	3040	3085	3130	45
962	3175	3220	3265	3310	3356	3401	3446	3491	3536	3581	45
963	3626	3671	3716	3762	3807	3852	3897	3942	3987	4032	45
964	4077	4122	4167	4212	4257	4302	4347	4392	4437	4482	45
965	98 4527	98 4572	98 4617	98 4662	98 4707	98 4752	98 4797	98 4842	98 4887	98 4932	45
966	4977	5022	5067	5112	5157	5202	5247	5292	5337	5382	45
967	5426	5471	5516	5561	5606	5651	5696	5741	5786	5830	45
968	5875	5920	5965	6010	6055	6100	6144	6189	6234	6279	45
969	6324	6369	6413	6458	6503	6548	6593	6637	6682	6727	45
970	98 6772	98 6817	98 6861	98 6906	98 6951	98 6996	98 7040	98 7085	98 7130	98 7175	45
971	7219	7264	7309	7353	7398	7443	7488	7532	7577	7622	45
972	7666	7711	7756	7800	7845	7890	7934	7979	8024	8068	45
973	8113	8157	8202	8247	8291	8336	8381	8425	8470	8514	45
974	8559	8604	8648	8693	8737	8782	8826	8871	8916	8960	45
975	98 9005	98 9049	98 9094	98 9138	98 9183	98 9227	98 9272	98 9316	98 9361	98 9405	45
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979	0783	0827	0871	0916	0960	1004	1049	1093	1137	1182	44
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981	1669	1713	1758	1802	1846	1890	1935	1979	2023	2067	44
982	2111	2156	2200	2244	2288	2333	2377	2421	2465	2509	44
983	2554	2598	2642	2686	2730	2774	2819	2863	2907	2951	44
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985	99 3436	99 3480	99 3524	99 3568	99 3613	99 3657	99 3701	99 3745	99 3789	99 3833	44
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993	6949	6993	7037	7080	7124	7168	7212	7255	7299	7343	44
994	7386	7430	7474	7517	7561	7605	7648	7692	7736	7779	44
995	99 7823	99 7867	99 7910	99 7954	99 7998	99 8041	99 8085	99 8129	99 8172	99 8216	44
996	8259	8303	8347	8390	8434	8477	8521	8564	8608	8652	44
997	8695	8739	8782	8826	8869	8913	8956	9000	9043	9087	44
998	9131	9174	9218	9261	9305	9348	9392	9435	9479	9522	44
999	9565	9609	9652	9696	9739	9783	9826	9870	9913	9957	43
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